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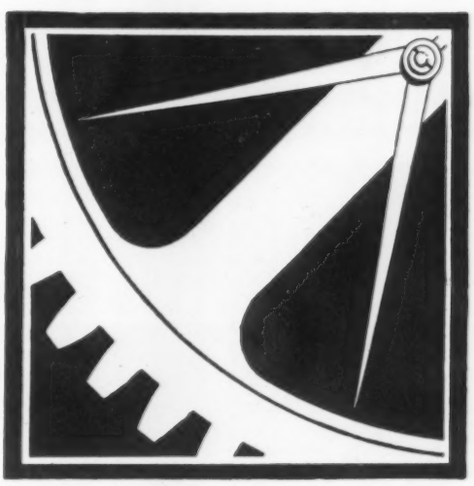
MACHINE DESIGN



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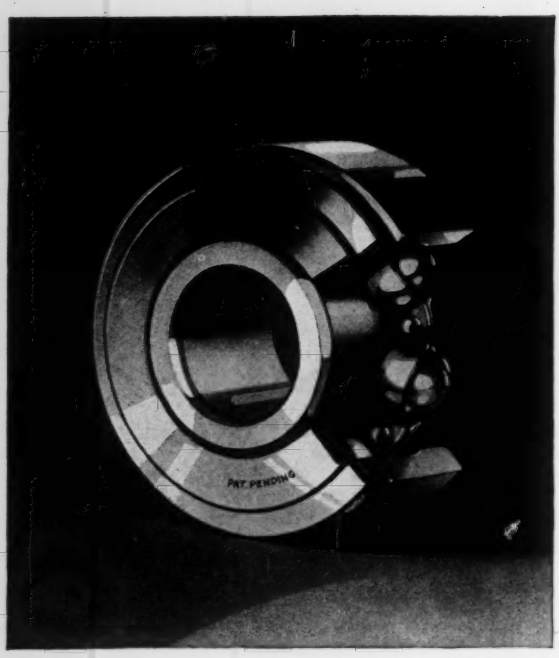
ENGINEERING–PRODUCTION–SALES



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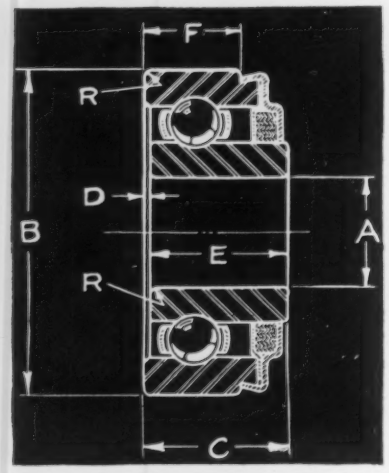

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GS-96	6	.2362	19	.7480	.3543	.016	.3383	.2362	1	.039
GS-97	7	.2756	22	.8661	.4060	.016	.3900	.2756	1	.039
GS-98	8	.3150	22	.8661	.4060	.016	.3900	.2756	1	.039
GS-98246	6	.2362	24	.9449	.4060	.016	.3900	.2756	1	.039
GS-98247	7	.2756	24	.9449	.4060	.016	.3900	.2756	1	.039
GS-9824	8	.3150	24	.9449	.4060	.016	.3900	.2756	1	.039
GS-99	9	.3543	26	1.0236	.4527	.016	.4367	.3150	1	.039
GS-200	10	.3937	30	1.1811	.5118	.016	.4958	.3543	1	.039
GS-201	12	.4724	32	1.2598	.5512	.016	.5352	.3937	1	.039
GS-202	15	.5905	35	1.3780	.5905	.016	.5745	.4330	1	.039
GS-203	17	.6693	40	1.5748	.6693	.016	.6533	.4724	1	.039
GS-204	20	.7874	47	1.8504	.7480	.016	.7320	.5512	1	.039
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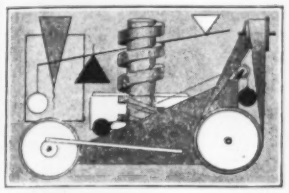
ENGINEERING–PRODUCTION–SALES

Volume 2 | June, 1930 | Number 6



Next MONTH

REDUCTION in weight not only is necessary in aircraft but in innumerable other types of equipment. For this reason an article to be included in the next issue on designing to reduce weight, prepared by the assistant chief engineer of one of the leading aircraft engine builders, will prove invaluable to those members of the profession confronted with weight-strength problems.

Application of steel forgings in design of machinery has been touched upon briefly from time to time in earlier issues. Of more comprehensive character is the series of articles on this subject to appear shortly.

L. E. Jermy
Managing Editor

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Calendar of Meetings and Expositions

June 16-19—American Society of Agricultural Engineers. Twenty-fourth annual convention will be held at Moline, Ill., with the LeClaire hotel as headquarters. There will be four morning sessions and the annual business session will be held on the evening of June 17. The afternoons of June 16 and 17 will be devoted to demonstrations and the two remaining afternoons to visits to factories and sight-seeing. At the banquet Wednesday evening June 18, L. J. Fletcher will be toastmaster. Among the highlights of the program will be, "Training Men for Research in Agriculture," by J. B. Davidson, Iowa State college; "Scientific Research in Agriculture," by Dr. A. F. Woods, director of scientific work of the department of agriculture; "A Review of Agricultural Engineering Research," R. W. Trullinger, senior agricultural engineer, office of experiment stations, department of agriculture, and "The World Engineering Congress," by H. B. Walker, agricultural engineer, University of California.

June 16-20—Association of Iron and Steel Electrical Engineers. Annual meeting of the organization to be held in conjunction with the iron and steel exposition at Buffalo. Technical sessions will be carried on at Statler hotel, and Broadway auditorium will be the scene of the exposition. Included in the program announced for the meeting are papers covering electrical developments and shop practices in the iron and steel industry, antifriction bearings for mill-type motors, general purpose motors and electric overhead traveling cranes. A special feature of the morning session on June 18 will be a discussion of antifriction bearings for roll necks by steel mill executives, engineers and antifriction bearing manufacturers. Latest developments in wire drawing and wire forming machinery also will be discussed. At Broadway auditorium 50,000 square feet will be devoted to all latest type apparatus used in producing iron and steel. Wire making and welding pipe electrically will be features of the exposition. John F. Kelly, 1010 Empire building, Pittsburgh, is managing director.

June 16-25—Second Plenary World Power Conference. To be held at Berlin, Germany. O. C. Merrill, Edmunds building, Washington, is chairman of the American committee.

June 18-25—American Railway association. Annual meeting of the mechanical division at the new auditorium, Atlantic City, N. J. During the same period an exposition of railway supplies and equipment will be held under the auspices of the Railway Supply Manufacturers' association. V. R. Hawthorne, 431 South Dearborn street, Chicago, is secretary of the American Railway association and J. D. Conway, 1841 Oliver building, Pittsburgh is secretary of the Railway Supply Manufacturers' association.

June 23-27—American Institute of Electrical Engineers. Annual summer convention to be held at Royal York hotel, Toronto, Ont., Canada. F. L. Hutchinson, 33 W. Thirty-ninth street, is secretary.

June 23-27—American Society for Testing Materials. Thirty-third annual meeting to be held at Chalfonte-Haddon Hall, Atlantic City, N. J. The technical program includes a symposium on aircraft materials used not only for structural members but also the materials entering into aircraft engines, including engine failures and causes thereof. Several papers deal with corrosion, fatigue, and corrosion-fatigue of metals. In addition a report is being presented summarizing the progress during the year in fatigue testing. The society's committee A-10 on iron-chromium, iron-chromium-nickel and related alloys plans to bring up to date the data on corrosion-resistant alloys presented in connection with the symposium held in 1924. The nonferrous metal field also will be covered by papers on various interesting subjects. The Charles B. Dudley medal will be awarded to J. R. Townsend, W. A. Straw and C. H. Davis. C. L. Warwick, 1315 Spruce street, Philadelphia, is secretary of the society.

June 26-28—Steel Founders' Society of America. Semi-annual meeting at Greenbrier hotel, White Sulphur Springs, West Va. The program includes a discussion on "Economic Co-operation," by Charles F. Abbott, executive director of the American Institute of Steel Construction; a talk on important conditions in the industry and the methods of meeting them by E. St. Elmo Lewis, Detroit; and other addresses by Dr. Hugh P. Baker, of the chamber of commerce of the United States, Roger L. Wensley, president of G. M. Basford & Co., New York and J. Nobel Borden, secretary of the American Arbitration association. G. P. Rogers, 932 Graybar building, New York, is managing director.

Aug. 31-Sept. 7—International Congress for General Mechanics. Meeting in Liege, Belgium. For information address Alb. Schlag, 4 Place Saitn-Lambert, Liege, Belgium.

Sept. 22-26—American Society for Steel Treating. Annual meeting to be held in connection with the National Metal exposition at Stevens hotel, Chicago. W. H. Eisenman, 7016 Euclid avenue, Cleveland, is secretary of the society.

MACHINE DESIGN
June 1930

Combating Vibration in Design of Centrifugal Machinery

By L. C. Schenck

INTERESTING and suggestive from its discussion of problems overcome in the design of a centrifugal extractor, and timely from its treatment of the subject of operating difficulties, the accompanying article will be found extremely valuable. The author is chief engineer with the Troy Laundry Machinery Co. Inc., East Moline, Ill.

WHILE laundry machinery occupies a strictly defined province of its own, the combinations of mechanical principles used therein can be applied in so many other fields that a concrete illustration of human-factor-controlled design of such a machine may promote this design policy in other lines.

A prominent example may be found in the modern centrifugal extractor used to extract the bulk of water from fabrics after their removal from the washing machine, thereby leaving them just damp enough to permit their being ironed or pressed immediately without need of other drying operation. The principle of centrifugal extraction is old in many other industries also, having been used for years for extracting oil from metal turnings and chips, extracting the liquids from fruits and roots, clarifying petroleum, etc., but in its adaptation to the handling of a family laundry bundle haye arisen distinc-

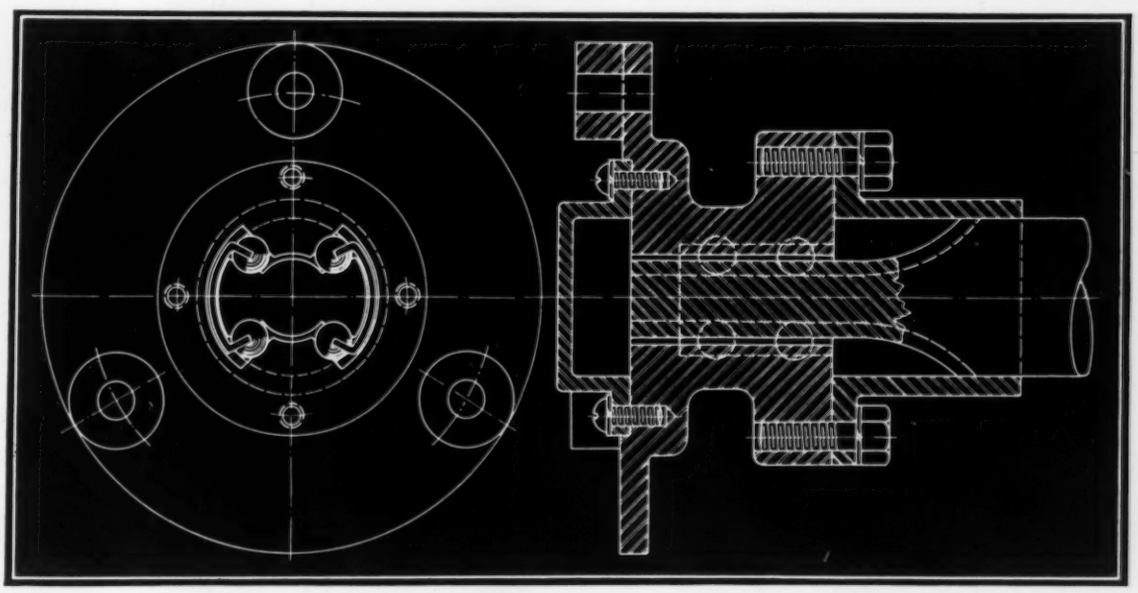


Fig. 1—Antifriction spline developed to overcome binding between spline members

tive problems which have required many years to solve ideally. As usual, however, the solutions were simple, once they were discovered.

Figs. 3 and 4 show front and rear views respectively of one of the most modern types of laundry extractors. In Fig. 2 will be seen a sectional view.

Every laundry extractor is composed of three principal units: the basket or internal container which receives the garments or fabric to be dried; the curb, or external shell which serves as a mounting for the basket and acts as a receptacle for the water extracted; the drive unit by which the basket is rotated at a high rate of speed.

Extractors are built with baskets of various diameters, ranging from 18 inches to 72 inches. The illustrations show one having a basket 48 inches in diameter and 21 inches deep, which is the most popular size. The basket bottom, or flange, is of cast iron covered with a spun sheet of tinned copper. The side walls are of perforated heavy tinned copper, reinforced by endless steel bands. The top reinforcing ring is of tinned bronze.

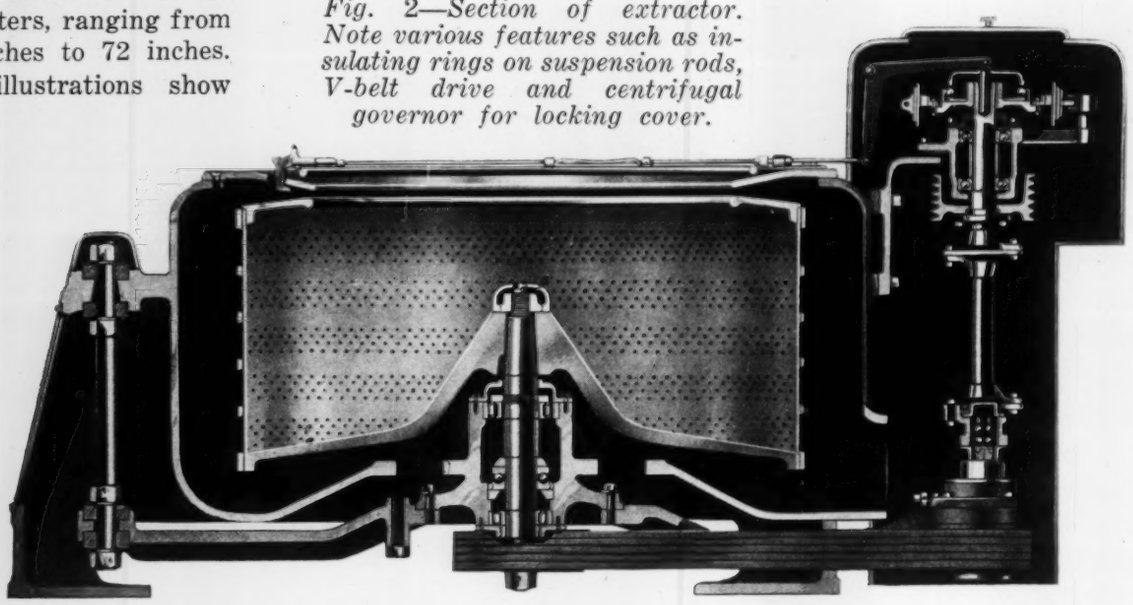


Fig. 2—Section of extractor. Note various features such as insulating rings on suspension rods, V-belt drive and centrifugal governor for locking cover.

Design Must Be Studied Carefully

When it is considered that this basket rotates at 800 revolutions per minute and that the load of fabrics when wet weighs over 900 pounds one can realize the destructive force pent up in this machine and the precautions which must be observed in its design and construction. This centrifugal force, of course, means mainly the application of suitable metal mass where needed.

The major problem, however, is that of satisfactorily dissipating vibration before it is transmitted to foundation and building. It is obvious that it would be practically impossible to place a load of wet fabrics in the basket without throwing it more or less out of balance, and in the case of extracting family bundles in large nets or bags, even distribution of the load becomes so much more difficult. While the basket is rotating with such a load, its tendency is to oscillate or waltz about its true center of gyration rather than the spindle center, and the greater the force restraining it from oscillating, the greater will be the abuse on spindle bearings and foundation.

Before the advent of the machine shown, the common method of dissipating this vibration in under-driven machines of this size (basket drive below basket) was to mount rigidly the spindle bearings in the bottom flange of the curb itself, and to suspend the entire body of the machine from three supporting posts by means of three steel rods having a ball universal joint at each end. In this type of machine, a motor having a vertical spindle was rigidly bolted to the side of the curb or shell and was belted direct to the spindle.

A machine of this type requires fewer parts than the construction illustrated but this saving is effected at a great sacrifice in operating characteristics. It will be seen that with this arrangement the entire curb and motor, in fact, every part of the machine except suspension rods and posts, was compelled to oscillate or vibrate with the basket upon every basket revolution. To say nothing of the power required to oscillate this entire mass, and ignoring the abuse imposed on basket spindle bearings which were restrained from movement by the weight of curb and motor, the most important objection after all was the psychological effect upon the operator.

Picture if you can an enormous mass of machinery weighing about two tons, shaking itself at a high frequency, rattling at every loose joint, pounding terrifically if there exists any wear or

clearance in spindle bearings. And it was not only the nerve-jarring caused by the noise alone to which the operator was being subjected, but the fact that as he looked at that vibrating, rattling monster he constantly was being reminded of the great destructive force confined therein, a potential hazard to his own life and the lives of his co-workers. Obviously, if the human factor deserved any consideration, here was a problem worthy of the designer's noblest endeavors.

Over sixty years ago efforts were directed toward the perfection of a self-balancing extractor, which term does not mean that the machine's load re-adjusts itself to a perfect balance, as the expression implies, but refers to the method of permitting the basket and spindle to oscillate independently of the machine's remaining parts thereby dissipating the vibration at the source of its generation. While these efforts were successful in over-driven extractors, difficulties arose in loading and unloading.

In the clear top or under-driven type of machine, since the spindle could not project above the basket, it was of course, impossible to provide it with a fulcrum bearing. The next step, therefore, was to mount the basket and spindle bearing unit upon a relatively light steel spider having three projecting arms which could be suspended from a stationary curb as shown in Fig. 2. This decreased to a minimum the number of parts which must oscillate, also the amount of weight restraining the basket swing. Furthermore, with suspension rods of the great length shown, the restraining effect of gravity was found to be negligible. The rods were insulated from curb and spider by rubber rings as may be seen in Figs. 2 and 5.

Fig. 4—Rear view of machine showing discharge port, vertical motor and projecting arms on inside of which suspension rods are hung

The next problem concerned the method of driving this basket, which could swing independently of the stationary curb, when at the same time it was desirable to keep the motor stationary by mounting it on the curb. It seemed as though it would be impossible to connect the stationary motor with the swinging basket without interfering with the basket's balance, such as is the case when a spinning top is touched by a finger.

The ideal solution which was finally conceived is clearly indicated in Figs. 2 and 5.

On a fourth arm extending from the basket spider was mounted a ball bearing shaft unit, this shaft being connected to the basket spindle by means of V-belts and sheaves. Since these two centers were tied rigidly together, the belt pull could not throw the basket off center.

A similar ball bearing shaft unit then was mounted on the stationary curb immediately above the first, and the two shaft units were connected by an automobile type of drive shaft having two universal couplings. The stationary upper shaft was provided with a V-belt drive extending to the curb-mounted motor.

Inasmuch as a secondary unbalance of the basket would result in a slight vertical motion or vibration of the spider drive shaft, and since if not dissipated close to the spider this vibration would be transmitted to the upper drive shaft and thence to the

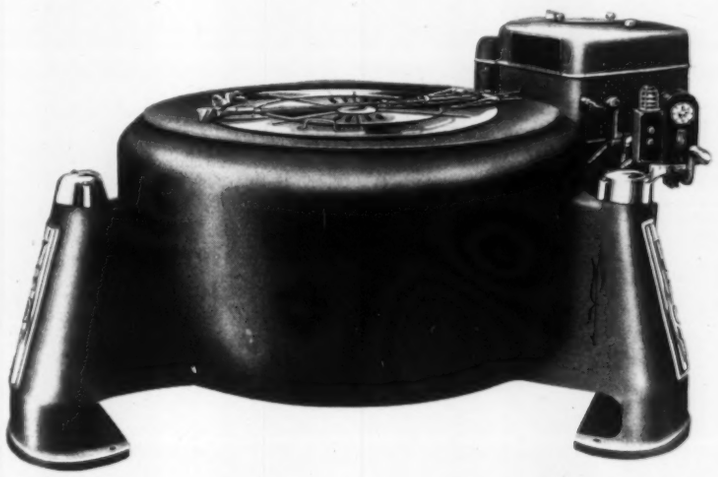


Fig. 3—Front view of extractor showing timer at right

curb, an ordinary automobile spline was at first interposed between the lower drive shaft and the universal shaft. In operation, however, it was found that the friction between male and female spline parts was so great as a result of the accelerating drive torque, that the two parts practically froze together and the vibration was carried to the curb regardless.

Why not an antifriction spline? Since no such thing could be found in commercial use, it was necessary to develop one for this application and it proved to be the difference between the machine's success and failure. As shown in Figs. 1 and 2, the hardened steel spline parts were provided with four sets of raceways, in each of which reposed two steel balls which acted as driving keys, at the same time permitting a frictionless relative vertical motion of the two spline members. As shown, two hardened steel retainers were used to keep the balls separated.

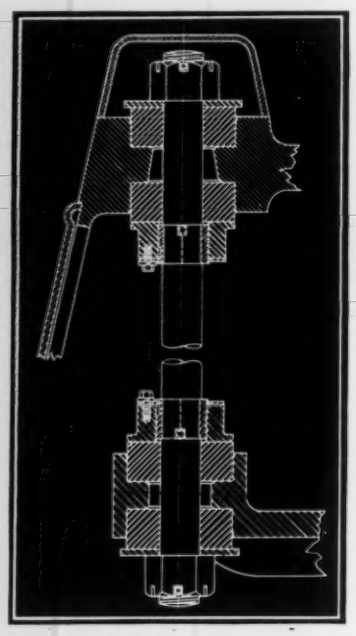


Fig. 5—Section of suspension rod. Instead of threading rod for inside adjusting nuts, at which point a V-thread would contribute to fatigue failure, a filletted shoulder is provided on which rests a split threaded sleeve. This is prevented from turning by milled notches engaging a pin which projects ¼-inch into the suspension rod

Human Factor Again Is Regarded

It next was necessary to provide a brake which would stop the basket within a reasonable braking period. At first thought the logical place for the brake drum would be directly upon the lower basket flange, which construction was used on the older type of extractors. Here again heed was given to the human factor. With the brake so mounted it would be necessary in the case of maintenance or relining of the brake to open the curb and remove the entire basket. It was considered that the V-belt capacity was far in excess of that needed during acceleration, and that if the brake were mounted on the upper drive shaft where it would be easily accessible the braking duty on the belts would hardly exceed the accelerating requirements. Therefore this construction was adopted and proved highly satisfactory in service.

An interesting feature of the machine is its safety cover over the basket provided for the operator's protection. After the basket has been loaded it is necessary to close the cover before power can be applied to the motor. When the cover is lowered it is locked automatically in place by a spring actuated latch shown in Fig. 2 at the left side of the cover. This latch is connected by a rod to a bell crank which supports a cap-shaped casting as shown. Below this cap are provided two inclined raceways attached to the brake drum and in these raceways two steel balls loosely repose. Since before the latch can be withdrawn the cap must be free to drop to the raceways, it will be seen that it is impossible to unlock the cover until the basket has come to a dead stop, since the balls are thrown out to the position shown and provide interference with the cap's movement. The result is a simple centrifugal governor safety device having no bearings or joints to stick or require lubrication.

It also is interesting to note that the machine's operation after being started is fully automatic, another thought for the operator's productive efficiency. After loading the basket he sets the timer shown in Fig. 3 for the desired running period, raises the lever immediately beneath it and presses the starting push button. The machine needs no further attention until ten minutes later, at which time the motor shuts off itself, the brake is applied automatically, and a signal light informs the operator that the machine is ready for another load.

Maintenance requirement has been reduced to a minimum in the design of the extractor. Anti-

(*Concluded on Page* 41)

YOUR articles dealing with consideration of the human element in design of machinery convey a message which, in its importance to manufacturers seeking the road to success, is limited not only to those machines with which we would first associate the idea, such as automobiles and the many other instruments of everyday living, but which well may be carried into remote manufacturing corners and will bring abundant reward to those who heed it.—L. C. SCHENCK.

SCANNING THE FIELD FOR IDEAS

A Monthly Digest of New Machinery, Materials, Parts and Processes, with Special Attention to Significant Features and Trends in Design

RESEARCH in the field of light alloys is proceeding rapidly. Many of the alloys introduced recently have numerous potential possibilities for application in the design of machinery.

One of the interesting examples is that of an alloy termed Hiduminium R. R. 50, discussed in a paper presented at a recent meeting of the British Institution of Production Engineers, which has been produced through the efforts of the Rolls Royce research laboratory in England. This alloy was used for the crankcase, cylinder block and head of the engine of the plane which won the Schneider Trophy last year and is used for all light alloy castings in aero engines and cars manufactured by the company. It is believed also that forgings now made of other aluminum alloys will in the future be made of Hiduminium. Composition of the new alloy is:

Copper	0.5 to 5.0
Silicon	0.2 to 5.0
Iron	0.6 to 1.5
Magnesium	0.1 to 5.0
Nickel	0.2 to 1.5
Titanium	Below 0.5
Aluminum	Remainder

According to the author of the paper, casting and forging of the material undoubtedly is easier than is the case with any other aluminum alloy. All grades are capable of being strengthened by heat treatment, and as the temperature in the case of castings does not exceed 175 degrees Cent., distortion or cracking during quenching operations is obviated.

Another alloy around which much interest centers is magnesium. This has a specific gravity lower than that of the high aluminum alloys, with greater tensile strength. Development and widespread employment of the alloy has been somewhat slow on account of its high cost and the difficulty of working it; however, castings of 100 pounds and over already have been produced successfully and it is predicted by numerous authorities that its use will increase where particularly high strength combined with lightness is essential.

Among the non-metallic materials introduced recently is Linatex, a new form of cold-cured rubber marketed by Wilkinson Process Rubber Sales Corp., Chicago. It is known as a non-ageing rubber, accelerated laboratory tests having shown that its life is at least four to five times that of commercial vulcanized rubber.

Abrasion resistant qualities of this new material render it particularly suitable as a lining for launders and chutes, hydraulic hoists handling sand and gravel, and for pump runners dealing with similar materials. It also has excellent properties as a vibration dampener, and should find

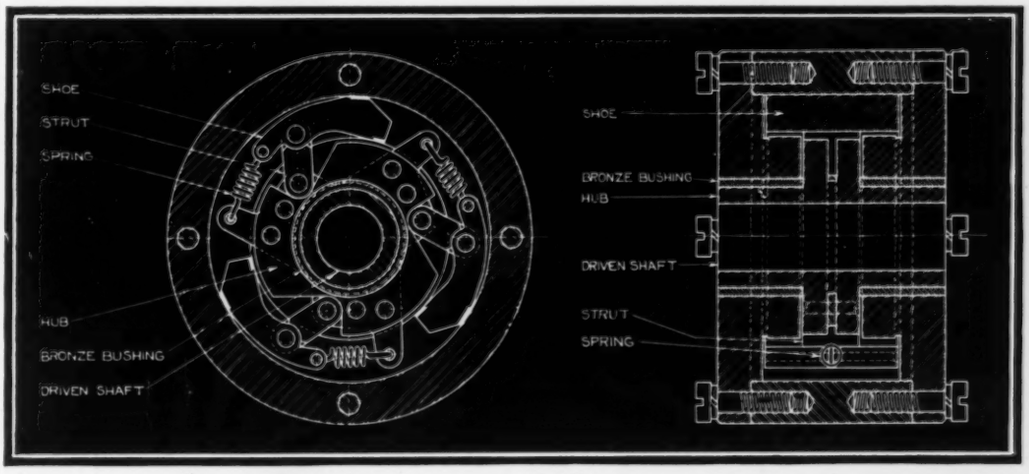


Fig. 1—Diagram showing design and general construction of the Pitter one-way clutch described on the following page

many applications in design, particularly where conditions of this nature are present or where a non-abrasive material is essential.

One-Way Clutch Replaces Ratchets

A ONE-WAY clutch possessing the feature of being instantaneous in clutching and release has been placed on the market by Universal Gear

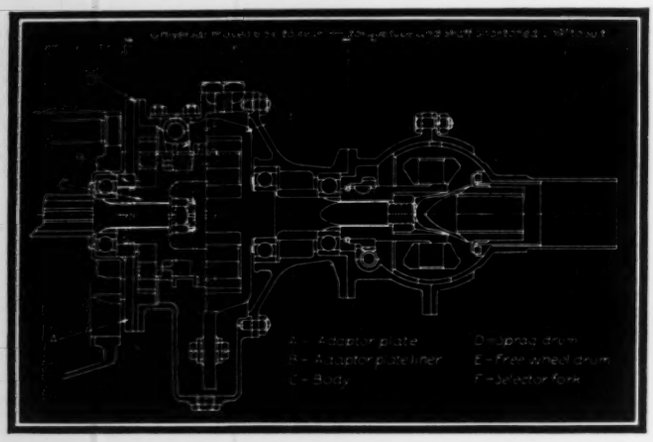


Fig. 2—Installation diagram of free-wheel

Corp., Chicago. This clutch emphasizes the suitability of wedging action as a means for locking sliding surfaces together, and serves as an indication that if properly applied, such action will give a positive lock without jamming.

Fig. 1 shows two sectional views of the clutch. It will be seen that a central spindle is provided, carrying three pivoted arms. On the end of each is assembled a clutch shoe. Coil springs keep the shoes in place for gripping between the inner surface of the driving member and the outer surface of the hub attached to the driven shaft. When the outer member of the clutch is rotated in the driving direction, each of the shoes immediately locks on two points between the driving and driven members.

According to the manufacturer, rapidity of action is practically unlimited. Some clutches now are in operation which lock and unlock thirty or more times per second. All moving parts of the clutch operate in a bath of oil and the case is made oiltight to eliminate the necessity for renewal of lubricant.

Will Cars Use Free-Wheel Device?

AN AUTOMOBILE coupling having many possibilities and embodying numerous interesting design features recently has been taken over as far as manufacturing rights are concerned, by Armstrong, Whitworth & Co. Ltd., Newcastle, England. It has been developed by Frictionless Mastagear, Ltd. and is known as the Millam free-wheel and sprag.

The more important details of the unit are shown in Fig. 3 and an installation diagram is given in Fig. 2. Insertion of the unit in the propeller shaft of automobiles is claimed to give the following advantages. When the foot is taken off the accelerator the engine revolutions die away to "ticking-over" speed and the car coasts along. The load is taken up when accelerator again is pressed down. In changing gear it is not necessary to disengage the clutch and any gear can be selected with ease, irrespective of the car speed.

This gear is in reality an automatic metal to metal clutch, the drive being taken between the outside faces of six segments and the inside surface of the driven drum. The rollers are not called upon to take the drive between the driving and driven members, which would give line contact only with consequently limited life of the parts involved and the possibility of slippage.

Radial Thrust Is Balanced

The driving portion of the free-wheel consists of a cage provided with six jaws, between each of which lies a pair of rollers. On the outside of the rollers, and immediately inside the drum, are six driving segments, shaped on their inner faces to form double-acting wedges. On the inside of the rollers is a hardened steel roller path keyed to a boss inside the driven drum. As this roller path is integral with the driven drum the radial thrust necessary on the segments to enable them to take the drive is balanced within the one driven member. Therefore there is neither radial thrust nor

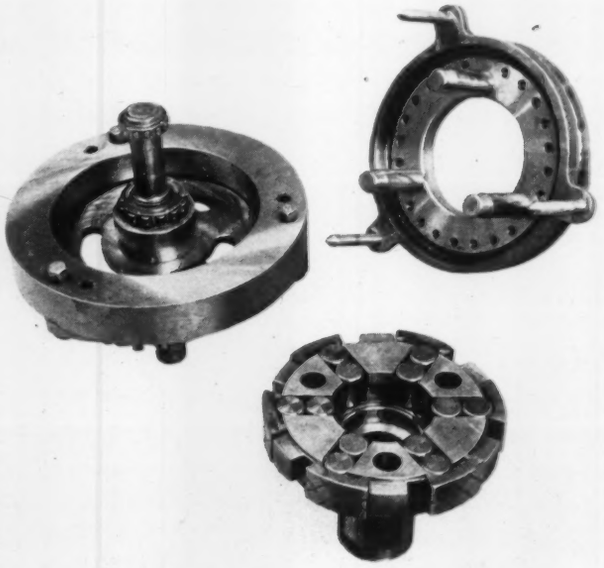


Fig. 3—Elements of the auto free-wheel and sprag by the use of which gears may be changed without declutching

end thrust between the driving and driven portions of the clutch. No slip can take place no matter what torque is transmitted, providing the angle of the wedge surfaces of the segments is less than the angle of friction between the segments and the drum. This angle has been determined after extended tests with various materials for drum liner and segments, using numerous kinds of lubricating oil.

Rollers Are Caused to Rotate

Another function of this fixed roller path is to cause the rollers to rotate slightly, due to the momentary relative rotary movement of the roller cage and roller path when the load is on the point of being taken up. This slight rotation of the rollers causes the outer rollers to roll along the wedge surfaces on the inner faces of the driving segments and helps the action of the drum in dragging the segments over the rollers. Practically instantaneous take-up of the load thus is assured at all times.

When the mechanism is used as an over-running device, the wedges are restrained from rotary movement in one direction relatively to the cage by means of axially sliding fingers, so that no wedging action can take place when the car overruns the engine. When, however, it is desired to use the mechanism as a solid coupling the restraining fingers are withdrawn. Wedging action then takes place in both directions of rotation and the necessary automatic pressure is put upon the driving segments, whether the engine is driving the vehicle forward, is being used as a brake or is driving the vehicle in reverse. The six axially sliding fingers are in the form of prongs on the side of the grooved controlling sleeve.

Action of the spragging mechanism of the device is similar, the equivalent of the driven portion in this case being fixed non-rotationally to the chassis of the vehicle. In the case of the sprag a somewhat different arrangement of rollers is employed, a single roller only being used in conjunction with each wedge segment, three of which are provided.

Locomotive To Yield Test Data

It would be hard to conceive a more thorough method of obtaining practical test data than that adopted by engineers of the Timken Roller Bearing Co., Canton, Ohio. Interest in the possibilities of antifriction bearings as applied to the axle journals of steam locomotives was responsible largely for their plan. Under the scheme the American Locomotive Co. was commissioned to build a locomotive to the specifications of Timken engineers, and when completed the engine was to be loaned to different railroads to be used in their regular service in order that bearing performance data could be obtained under every kind of operating condition. The locomotive passed its preliminary trials some time ago and was handed over to the Pennsylvania railroad for its first test late in May.

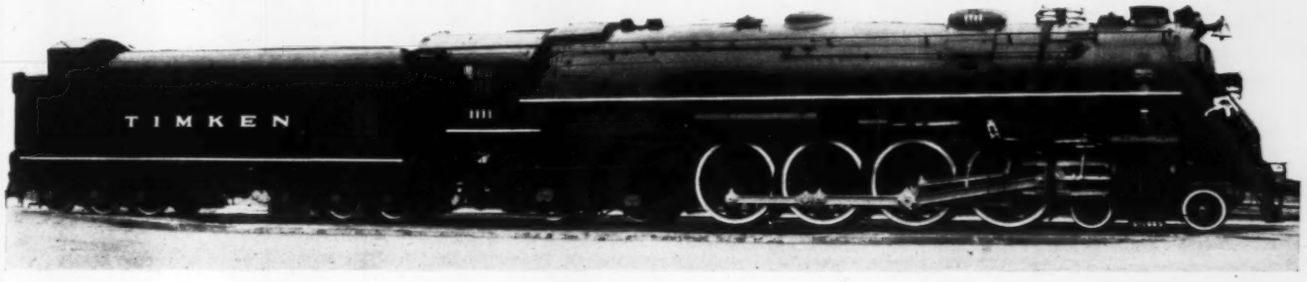


Fig. 4—Antifriction bearings have been applied to all main bearings of this locomotive, and alloy steels have been used in many instances, for obtaining practical test data

In the design of the locomotive, shown in Fig. 4, roller bearings have been applied to the main drivers, engine, trailer and tender trucks, and on the crankshaft and idler gear of the Franklin booster. Each application presented individual problems and operating requirements, therefore a different method of mounting the bearings was employed in each case.

Another feature of the locomotive that is of considerable interest is the use of alloy steel parts. A permissible top speed was obtained by the use of heat treated one-piece pistons, hollow piston rods, heat treated nickel vanadium steel cross heads, and side rods of normalized 2¾ per cent nickel steel. The result was that the weight of the reciprocating parts was reduced 460 pounds per side. In the case of the cross head guides, not only was weight reduced by use of alloy steel but wearing quality was improved. The steel is carburized and hardened to a degree that renders

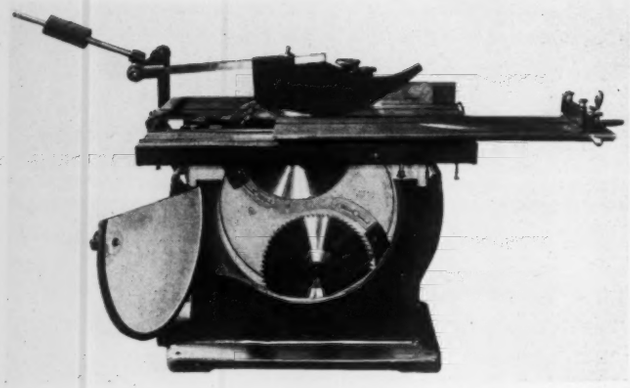


Fig. 5—Saw bench with individual motor driven saws on revolving yoke

it impervious to the action of dust or other substances likely to collect on the guides. Thus wear and also the hammering and knocking to which the driver bearings are subjected as a result of looseness in the cross head are reduced.

Motorized Rollers Used on Table

EMPLOYMENT of individual motors on machines has been increasing for many years. And now one can find instances where individual motors are employed for driving the numerous spindles of textile spinning frames and even for driving the individual rollers of a conveyor.

An application such as the latter, made recently by Reliance Electric & Engineering Co., Cleveland, is shown in Fig. 6. This illustrates a runout table from a hot mill stand. Obviously it is essential that the utmost precaution be taken to maintain uniform operation of the rolling mill and conveyor, as a breakdown or temporary stoppage would cause piling up of the heated metal. In the case of a serious stoppage this would be the means of causing a jam at the soaking pits or might even necessitate the damping down of open hearth furnaces. It is to cope with such serious consequences that conveyor table driving methods are being changed.

Earlier methods involved gang drives where the rollers usually were driven from a main motor through gears or rope connection. However, these gang drives have the disadvantage that trouble with one roller generally necessitates the shutdown of the whole table. On the other hand, motorized rollers still may act as conveyors in case of motor trouble and repairs can be made when convenient. The ratings of the induction motors used varies from one-fourth to five horsepower.

Saw Employs Dual Motor Drive

ANOTHER application of individual motors worthy of consideration wherever two or more operations are carried out on a single machine is embodied in the design of the universal bench saw developed by Oliver Machinery Co., Grand Rapids, Mich.

The machine, illustrated in Fig. 5, has two saw arbors, one carrying a rip saw and the other a cross cut saw. Each arbor is driven by a separate motor. The bearings for the arbor shafts and the individual motors driving them are mounted on a yoke which can be revolved readily when the saws either are stationary or in motion.

A hand wheel, the top of which is discernible in the illustration at the right of the body of machine, is employed for operating the yoke through a worm gear mechanism. For locking the yoke in desired operating position for either of the saws a somewhat novel method is employed, a clamp being provided which acts on

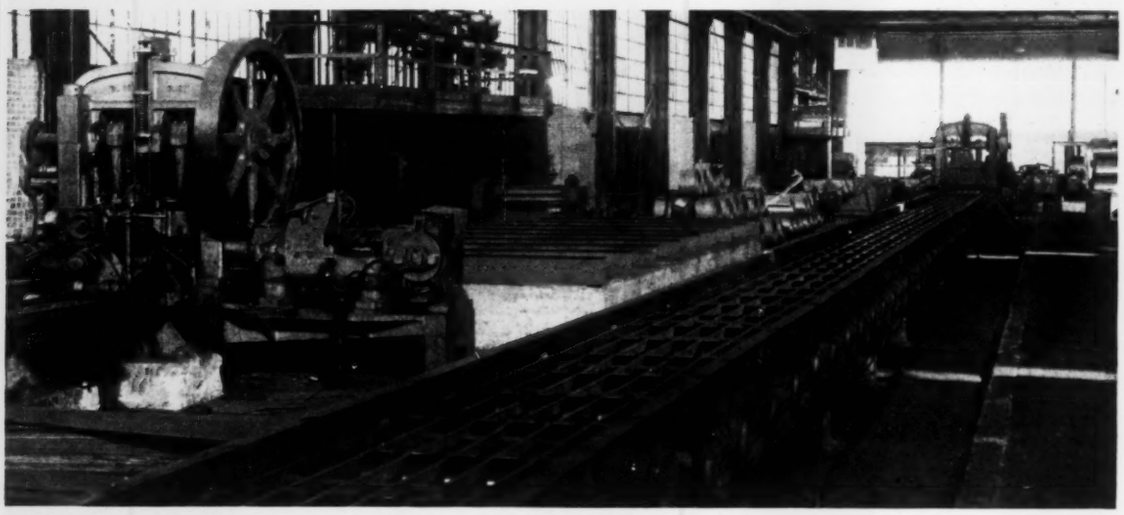


Fig. 6—Runout or conveyor table on which individual rollers are motor driven to guard against any possibility of shutdown

the worm shaft instead of the table mounting.

When either of the saws is not in use, the motor driving it stops automatically. In other words, as the yoke is revolved the motor which is approaching the operating position starts up and the other stops as it is being lowered into the bottom of the machine.

Contour Is Gaged by Laminations

AN IDEA possessing many possibilities in design work and which might be employed to advantage on testing machines is embodied in the "Maco" template introduced by the American Maco Template Co., New York. By its use, accurate profiles can be obtained in a few seconds.

The template, shown in Fig. 7, is composed of a number of fine strips of hard drawn solid brass, the thickness of each strip being .007 inch. Laminations are held in position by two clamps, and the laminations cannot fall out due to the special section of corrugated rubber with which the clamps are fitted.

To take an accurate outline of any contour it

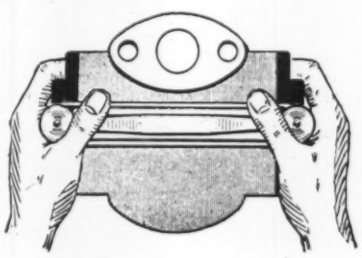


Fig. 7—Template built up of laminated plates by means of which contours are obtainable readily

is necessary to loosen the screws and press the adjustable instrument against the surface. The laminae arrange themselves to conform with the surface and when the pressure is complete the screws are locked to hold the plates in position.

Use of Carburetor Eliminated

SIMPLIFICATION of engine details is emphasized by the adoption of a device eliminating the usual carburetor and intake system of the gasoline engine and permitting the use of fuel ranging from furnace oil to high-test aviation gasoline, which recently has been perfected by the Pratt & Whitney Aircraft Co.

While the device has been developed from the point of view of the aviation engine, it is said to be equally adaptable to automobile or other gasoline engines. It has been used successfully for 6000 miles in an old model T Ford.

So complete is atomization that the highest and the lowest test fuel may be used. In tests, aviation gasoline was carried in one tank and furnace oil in another. While engaged in maneuvers, the pilot changed at will from one to the other tank.

Method Studied to Reduce Torsional Vibration

UNDAMPED torsional vibration of an engine shaft may cause serious damage to a machine. Therefore, reducing torsional vibrations by the use of a friction damper is a development of importance to the manufacturer and to the user of diesel and gas engines. This method of minimizing torsional vibration is used in the engines of the Graf Zeppelin. A specimen broken by undamped vibration is shown in Fig. 1.

Fig. 1—Specimen broken by undamped torsional vibration

The unit shown in Fig. 2 was developed by Dr. Den Hartog, research engineer, Westinghouse Electric & Mfg. Co. to study methods of decreasing torsional vibrations of shafts. In it, vibrations similar to those of an actual shaft in an internal combustion engine are set up, and a record of them is shown upon a screen by light reflected from a moving mirror actuated by the vibrations. The magnitude of the vibrations is noted readily by observing the path of the light upon the screen. At the critical speed this path is an elipse. Using a friction damper, torsional vibrations are reduced to a minimum, and reduction observed by movement of light beam.

Fig. 2—Unit developed to study method of reducing vibration

Last section of the series of articles dealing with a principle of increasing importance in the design of machinery

Hydraulics in Design

By John P. Ferris

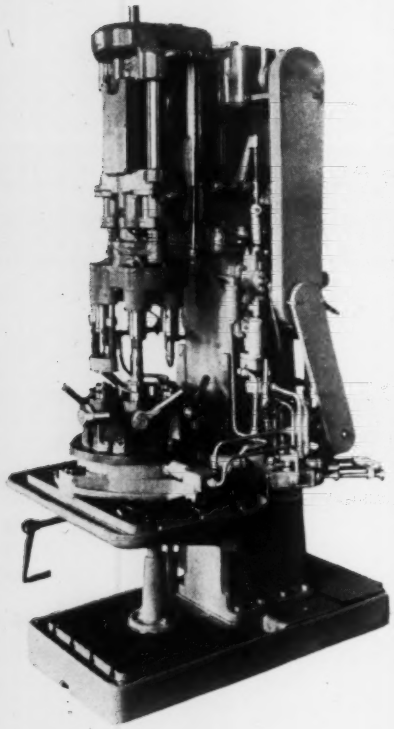


Fig. 1—Automatic drilling machine with hydraulic feed and hydraulic indexing of table

SOME uses of hydraulics were described in the second article of this series in the May issue, illustrating its value as a means of obtaining control of speed together with substantially positive, or "metered," motion for either rotary or linear motion. Flexibility in the location of the units, flexibility in the characteristics of the motions obtained, and the great variety of functions obtainable are important properties of the hydraulic system. The three machines described below illustrate these features.

Fig. 4 shows a large special boring and facing machine on which the three feeding motions are obtained hydraulically. Three cylinders are located in line with the cutting reactions. Each cylinder is driven by a separate pump, as it is necessary to control each motion separately. The three pumps are grouped for convenient drive from a single motor. One motion is controlled by a lever mounted on the pump; the controls for the other two are located at the operator's station.

Facilitates Location of Units

The machine illustrated shows well what is meant by flexibility of location. Each unit is where it should be, and no difficult design work was required to get it there. It is generally cheaper to build a special machine of this type using hydraulic units than when the same motions are obtained by mechanical means.

The planer-shaper of Fig. 7, on page 37 of the May issue, replaces an older design using a large gear wheel and crank located in the bed. The crank dimensions formerly limited the maximum stroke obtainable, and made for a weak bed casting by taking up space which was needed for adequate ribbing. The flexibility of location of hydraulic units means on this machine that a rigid bed can be used, as the table is reciprocated by a cylinder which occupies a space about 5 inches square in the sectional plane, thus permitting the use of strong ribbing. The machine is built in various length for strokes up to 90 inches. Only the lengths of bed, table and cylinder must be changed for longer or shorter machines.

Contrast this with the difficulty of redesigning for different lengths on a crank operated machine. The pump is completely enclosed in the column, and the valve which starts, stops and reverses the table motion is located under the cover on the near side of the bed, immediately below the table trips. While normally used for planing, the machine can also be used for the surface grinding

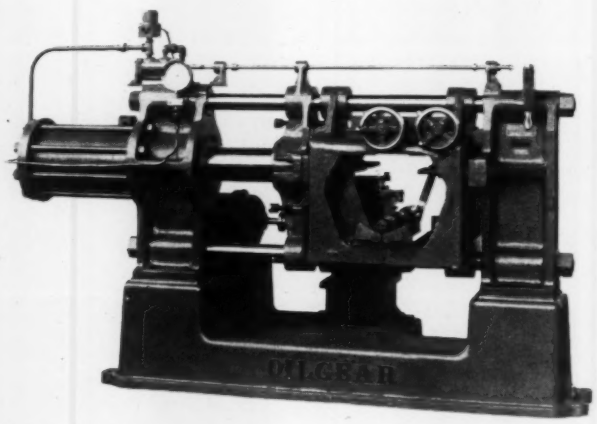


Fig. 2—Special machine for assembling bushings

of die blocks. There are two ranges of table return speed provided. For planing, the table returns at three times the forward speed and for grinding it returns at three-fourths of the cutting speed. This dual range of return speeds, in addition to the variation provided by changing the pump stroke, is obtained by shifting a valve; no additional mechanism is required. The length of stroke and the location of the reversal points of the table can be adjusted readily by locating the table trips wherever desired.

The cross feeding of the tool carrier also is accomplished hydraulically. At every table reversal the cross feed cylinder shown at the bottom of the vertical feeding shaft at the right side of the column makes a stroke and stops against an adjusting screw. The cylinder motion ratchets the feeding shaft; to vary the feed it is only necessary to adjust the stop which establishes the feeding cylinder travel.

In addition to flexibility of location of the hydraulic power and control units, the flexibility of location of the machine on which they are used is of great value. The welder shown in Fig. 4 on page 37 of the May issue and the straightening press shown in Fig. 11 on page 39 of the May issue can be moved around from one location to another in the user's shop as units, like machine tools. Both welding and straightening have been done hydraulically using System 3, in which the motor units take their power from a central accumulator system. When this is done, to move the machines is inconvenient and expensive; new pipe lines may be required and in any case repiping is necessary. An accumulator may not be available where they must be located. This drawback of the hydraulic method can be avoided by using an individual pump unit as an integral part of each machine.

THIS article concludes the three-part series by J. P. Ferris, chief engineer of the Oilgear Co., Milwaukee, on the principles of hydraulics and their application in design of machinery of various kinds. Further information on the advantages of this method of obtaining motion in machines is incorporated in this section of Mr. Ferris' series.

Fig. 1 shows a fully automatic drilling machine with a rotating table for carrying fixtures and work. The drill is fed and the table is indexed hydraulically. The example is given to show the possibilities of interlocking motions by the hydraulic method. This particular design is so arranged that the table must be completely indexed and an index pin driven home before it is possible for the drill to start down, to avoid any possibility of the drills striking the work or fixtures at an incorrect location. At the completion of the drilling operation, the downward moving drill spindle strikes a trip which reverses it automatically. Near the end of its upward stroke it actuates a valve which directs the oil flow to the indexing cylinder. After this cylinder has rotated the table to the desired position, the oil escapes through a port in the wall of the index cylinder and goes to a cylinder which pushes in the index pin. When this pin is all the way in, the oil escapes through a port in the side of the index pin cylinder and enters the main feeding cylinder, starting it down at rapid traverse speed. Just before the drills strike the work, a trip automatically reduces the downward velocity to the proper rate for feeding the drills. The cycle repeats indefinitely, the operator loading the fixtures at an idle station on the rotating work table.

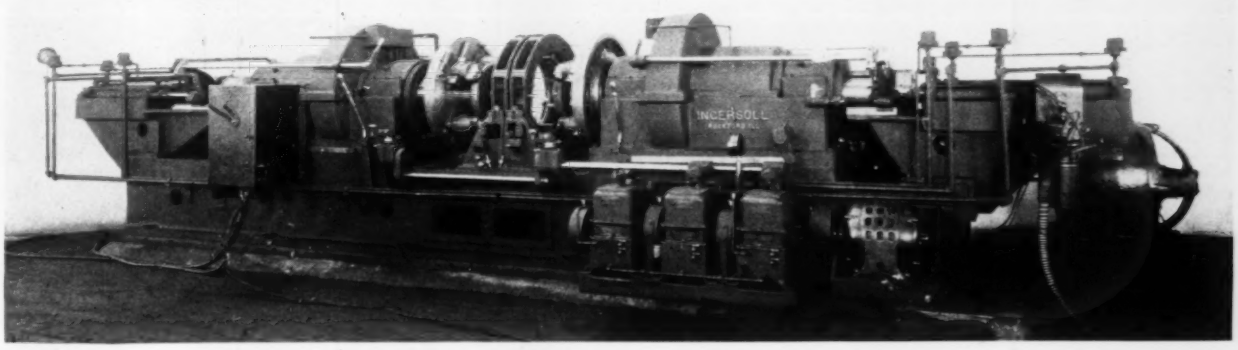


Fig. 4—Special boring and facing machine equipped with three hydraulic feeding motions

Flexibility Has Been Neglected

I have described a number of applications illustrating the versatility of the hydraulic method due to its features of flexibility of function and location, because I believe it to be one of the most important reasons for the present extensive use of hydraulic equipment on machinery. Many engineers fail to give due weight to the necessity of designing for flexibility. Processes are changing

rapidly today; the machines and methods which are best adapted to keep pace with the changes will become increasingly popular.

The hydraulic method makes possible an almost unlimited variety of functions. Pumps of differing displacements can be combined on the same job and controlled by shifting valves and other devices so that it is possible to obtain almost any desired speed ratio and sequence of motions. That an impressive variety of results can be obtained by combinations of fairly simple basic hydraulic units along lines well established in practice makes the method extremely attractive to the designer.

Of the sixteen principal properties of the system listed on pages 35 and 36 of the May issue, I have illustrated and discussed four important divisions. Limitations of space prevent any detailed discussion of the other twelve. The following brief remarks must serve to illustrate some of the more important points involved.

Fig. 2 shows a special assembling machine for pressing 16 valve guide bushings simultaneously into an automobile engine block. The hydraulic method provided:

1. Ability to locate the bushings accurately without any measurement, by stalling the parts of the special fixture against each other.
2. Short delivery on a very special piece of equipment. It was only necessary to build a suitable frame, a fixture, and to combine them with a standard pump unit, valve, cylinder, control and an ordinary induction motor.

Fig. 5 shows a 120 ton sheet metal blanking press designed for use with expensive dies. Hydraulics provided:

1. Ability to use a short stroke. The platen stalls against stops top and bottom, and can be run on a stroke as small as .005-inch if desired. One-sixteenth-inch is ample for the job. The upper die penetrates into the the lower die by only about .004-inch, thus limiting the amount of metal which must be ground away when sharpening the dies to a .004-inch layer. The consequent saving in die wastage is great.
2. Easy adjustment of the position of the stroke and of the ram speed. The ram speed is not affected by changes in length of stroke.
3. A marked decrease in rate of die wear due to "cushioning" effect of the fluid medium by which the force is applied.

On a turret lathe equipped with hydraulic feed, this method provided:

1. Ability to change the setup quickly, a feature of great advantage on short runs.
2. Ability to face a shoulder, stalling the carriage against a rigid stop while the cut cleans up, without over-straining the structure of the machine.
3. Ability to determine when the tools need regrinding by the use of a presure gage which gives a continuous indication of the load.

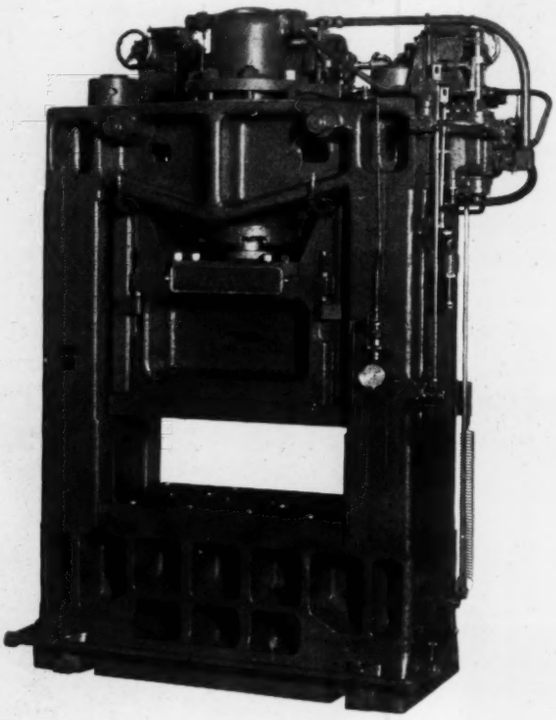


Fig. 5—120-ton sheet metal blanking press

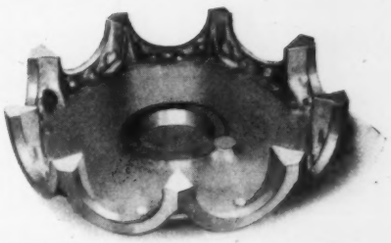
By the use of hydraulics on a surface grinder with a variable delivery pump and cylinder for reciprocating the table the following advantages were derived:

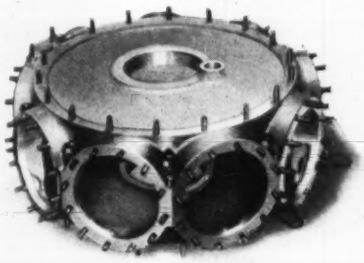
1. Increased production due to the high table speed which can be obtained without shock. The inertia of the moving mass is absorbed and dissipated harmlessly in the reversing valve.
2. A convenient solution to the important design problem of table reciprocation without the necessity of developing special equipment for the purpose. Standard units available on the market were used.

In these three articles I have attempted only to indicate the basic principles underlying the hydraulic art, and the directions in which to look for benefits from the use of hydraulics in design. Space conditions have permitted discussion of only some of the more important applications of hydraulics, and for further information the engineer must look to the leading builders of hydraulic equipment. Up-to-date books on the subject do no exist, so far as I know.

It is hoped that the tabulations of the basic hydraulic systems and their principal properties, together with the few examples given of their application in practice, will be of assistance in advancing the general knowledge of this art.

Applying Aluminum in the Wrought Condition





By F. V. Hartman

WROUGHT aluminum is produced in a number of different forms. As in casting practice, however, it generally is alloyed with other elements to improve its mechanical properties. Sheet is the most common form in which wrought aluminum is made; it may be rolled in any thickness from less than 0.0005-inch (aluminum foil) to 2 inches or more (aluminum plate). Aluminum is drawn into round, square, oval and special shaped seamless tubing. It also may be extruded; the extruded products include moulded parts, small structural shapes and many special shapes. Bar and rod sometimes are extruded but these more often are rolled; rolled stock includes most commercial shapes such as rounds, squares, and hexagons. Most of the common structural shapes also are rolled in sizes from a 6-inch channel to a 12-inch channel. Certain aluminum alloys can be forged readily. This process is used for connecting rods, airplane propellers and many types of levers and fittings.

NUMEROUS advantageous uses of wrought aluminum alloys are discussed in the accompanying article, the second of the two-part series by F. V. Hartman, Aluminum Co. of America. Important data for those directly concerned with design also are given.

Mechanical properties of some of the more commonly used wrought aluminum alloys are given in Table II.

Commercially pure aluminum and aluminum-manganese alloy containing 1.25 per cent manganese have been in common use for many years. These are not susceptible to heat treatment but are fabricated in various tempers based upon the properties which result from varying amounts of cold working done on the metal after it has been annealed; this applies particularly to rolled and drawn products but of course does not include extruded products which are more nearly in the annealed condition. The strong alloys of aluminum, a development of the last ten years, are heat treated to improve their mechanical properties. They may be had in a number of different tempers which are obtained by variations in the heat treatment to which the metal is subjected.

Commercially pure aluminum, designated 2S in the accompanying tables, is furnished in the annealed condition and in the quarter, half, three-quarter and full hard tempers; these are designated 2SO, 2S1/4H, 2S1/2H, 2S3/4H, and 2SH, respectively. This metal is worked easily and is well suited for drawing, stamping or spinning operations in which high strength is not necessarily of primary importance.

Alloy 3S is furnished in the same tempers as 2S, and these are designated 3SO, 3S1/4H, 3S1/2H, 3S3/4H, and 3SH. This alloy has appreciably higher mechanical properties than 2S, but

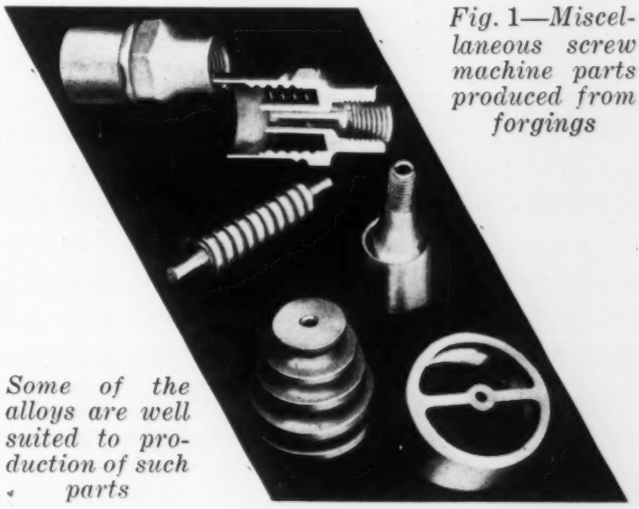


Fig. 1—Miscellaneous screw machine parts produced from forgings

Some of the alloys are well suited to production of such parts

can be worked almost as easily in corresponding tempers, and since it is resistant to corrosion, it is replacing 2S for many uses. Number 3S alloy is particularly well suited for equipment which must operate at elevated temperatures.

Alloys 2S and 3S are employed largely in the fabrication of tanks, large containers, pressure vessels and many types of special equipment used in the chemical and food industries. They generally are used in the form of sheet, tubing and extruded shapes.

The most commonly known strong aluminum alloy is the duralumin type, or 17S. This alloy is furnished in the annealed condition, 17SO, in the heat treated condition, 17ST, and sometimes in the hard rolled condition, 17SH. It is in its softest state when annealed and hence more workable. When formed in the annealed condition the forming operation should be followed by heat treatment to obtain the full mechanical properties. This alloy does not harden immediately after heat treatment; about four days are required for ageing to take place. However, during the first few hours after heat treatment the alloy is quite soft and easily worked. Advantage often is taken of this fact by forming the metal during this period; in such a case, the time interval should be as short as possible but may vary from one to four hours, depending upon the severity of the operation. Some of the less difficult forming operations can be done on 17S in the fully heat treated condition. Alloy 17SH is almost never used, although the metal in this condition sometimes is supplied to manufacturers who intend to do their own heat treating.

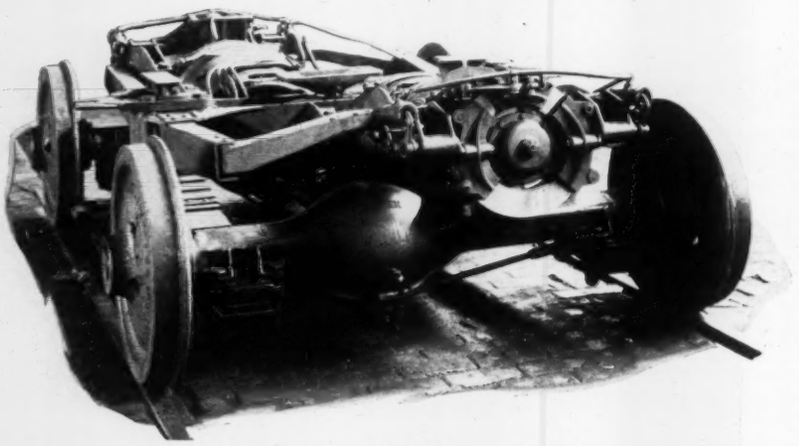


Fig. 2—Street car truck with frame built up largely of both wrought and cast aluminum alloys

Alloys Are Aged Artificially

The alloys, 25S and 51S, are furnished in the annealed and the heat treated conditions, the same as 17S alloy. Unlike 17S, however, 25S does not age spontaneously and only a partial ageing takes place in 51S following heat treatment. Both of these alloys must be aged artificially by reheating at a comparatively low temperature after heat treating to give them their fully heat treated properties. They therefore are available in an additional temper, designated 25SW and 51SW. These alloys in the "W" condition are more workable, particularly 51SW which can be formed easily.

Where highest strength and ductility together with maximum corrosion resistance are required, 17S alloy should be chosen. These properties qualify 17S particularly well for aircraft work. For the ordinary applications of the machine designer, however, the corrosion resistance of all the alloys in all tempers is satisfactory. Number 25ST alloy has practically the same mechanical strength as 17ST and can be formed or worked almost as easily. It forges readily and is the most commonly used aluminum forging alloy; 51S and 17S also can be forged when handled properly.

The ultimate strength of 51ST is considerably lower than that of 17ST and 25ST. However, 51ST is used widely owing to the fact that its yield point is practically the same as 17ST and 25ST and because it is more easily worked in the "W" condition than 25SW or 17S immediately after heat treating.

The alloys of aluminum have found considerable use in the transportation field for railroad car and bus construction where the economies of light weight are obvious. The bodies and some of the lifting mechanism of large motor trucks such as are used for hauling coal and gravel have been made economically of aluminum alloys; the maximum weight of such trucks is limited by legal restrictions and hence the increased cost of aluminum construction is soon paid for by the increased pay load. Considerable use is being made of aluminum alloys in elevator cage construction where better performance is obtained and economies are effected in reduced power cost, and smaller power unit together with lighter cables, counterweights, and sheaves. Aluminum construction lends itself readily to moving parts of machinery, especially those that move intermittently and must start and stop quickly. Forged aluminum connecting rods are being used successfully in different types of engines, particularly automobile motors.

The fact that aluminum alloys are non-magnetic makes them especially well suited for many applications in the electrical field such as relay

TABLE I

Composition of Wrought Aluminum Alloys

Aluminum Alloy No.	Approx. Compositions, Per cent Cu.	Mn.	Mg.	Si.	Al (min)
2S	....	....	....	....	99.0
3S	....	1.2	....	....	97.0
17S	4.0	0.5	0.5	....	92.0
25S	4.5	0.8	....	0.8	92.0
51S	....	....	0.6	1.0	96.5

covers, switch barriers, supports for various types of switches, motor brush holders, and brush holder fingers. In the latter case, in addition to being non-magnetic, aluminum alloy construction results in a lightweight finger which has less momentum and will, therefore, exert a more steady pressure against the brush under vibration.

Suggestions that were made for the design stresses and factors of safety in the article dealing with cast alloys which appeared in the April issue of MACHINE DESIGN, apply also to wrought alloys. In the earlier article it was stated that for impact loading the selection of a factor of safety should correspond to similar design practice for steel. The designer may be puzzled at this since laboratory impact tests, run on small notched specimens, give low values for high strength aluminum alloys when compared with steel. Comparative tests on frames of structural steel and similar ones of high strength aluminum alloy indicate, however, that each material will fail in about the same manner under the same impact blow.

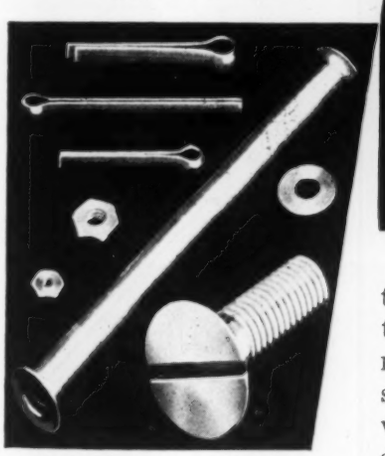
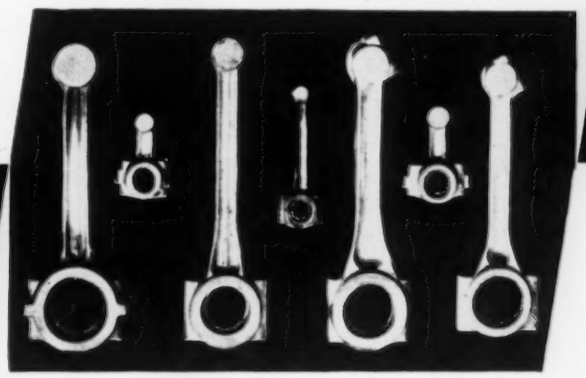


Fig. 3—(Below)—Other examples of screw machine parts and cotter pins fabricated from wrought aluminum



The effect of temperature on aluminum alloys is an important factor in design considerations. The same general conditions exist for wrought alloys as for cast alloys; the factors applying to the design stresses of some of the more commonly used wrought alloys are given in the Table III.

Aluminum structures such as tanks, containers and pressure vessels are usually of welded construction. In such cases, the strength of the structure is governed largely by the strength of the metal in or adjacent to the weld, even though material of an intermediate or hard temper is used throughout the job. In such a case it should be realized that the metal in the weld is in the cast condition while the metal close to the weld is in the annealed condition. If the welded joint is dressed down but not hammered, the properties of the cast material should be used in the selection of a design stress, since the properties of the metal in the as-cast condition are lower than when in the annealed wrought condition, corresponding to that which is adjacent to the weld.

In cylindrical containers subjected to internal presure, the stresses in longitudinal seams are double the stresses in circumferential seams. Advantage of this fact has been taken in the construction of small welded cylindrical tanks of heat treated aluminum alloys by constructing them with circumferential welds only. These tanks are heat treated after welding so that the metal in the completed tank is in the wrought and fully heat treated condition, except the metal in the weld which is in the cast condition and therefore has somewhat lower physical properties than it has in the wrought form.

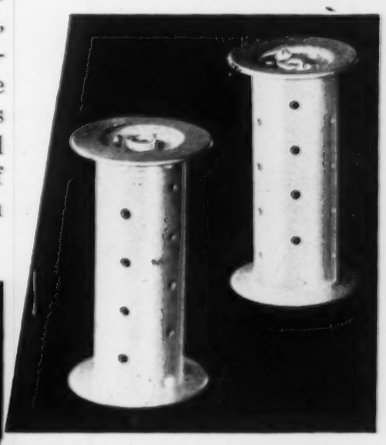


Fig. 4—(Above)—Many applications such as this can be made advantageously by use of alloys.

Fig. 5—(Left)—Forged alloy connecting rods

Stiffness Considerations

The modulus of elasticity of aluminum and its alloys is 10,000,000 pounds per square inch, while for steel it is 29,000,000. This means that aluminum will deform about three times as much as steel, provided the proportional limit is not ex-

ceeded. Since erroneous conclusions often are drawn from this fact, it seems well to point out that deformation is not a true comparison of strength between different materials. A ratio of

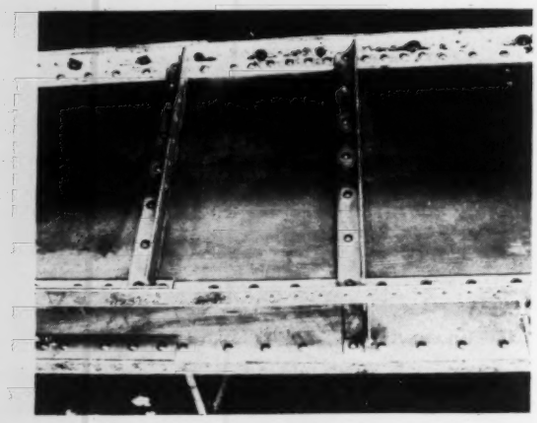


Fig. 6—Portion of aluminum box girder, showing riveted construction

three to one for deformation in steel and aluminum does not in any sense of the word mean that steel is three times as strong as aluminum. If we take, for example, a beam of 17ST and one of structural steel, both identical in section and similarly loaded, the 17ST beam will deflect about three times as far as the steel beam; nevertheless both beams will have about the same strength since the alloy 17ST and ordinary structural steel both possess approximately the same strength.

It sometimes is necessary to design an aluminum structure so that it does not deflect any more than a similar structure of steel. In such cases, it may be necessary to work at lower unit stresses and to increase the depth of beams in the aluminum structure. Take, for example, the deflection of a beam—this varies inversely as the product of the modulus of elasticity and the amount of inertia, or

$$d = \frac{K}{EI}$$

where K = a constant depending on the method of loading, E = modulus of elasticity and I = moment of inertia. From this it follows that for equal deflections, since the modulus of elasticity of steel is 2.9 times that of aluminum, the moment of inertia of the aluminum beam must be 2.9 times that of the steel beam. If we consider, for example, a beam of rectangular cross-section, it will be found that the depth of the aluminum beam must be 43 per cent greater than that of the steel beam, assuming that both beams have the same width. Nevertheless, the weight of the aluminum beam will be only approximately one-half that of the steel beam.

Riveting May Be Employed

Riveting is a very common form of fastening that is readily applicable to aluminum and aluminum alloys. In general, riveted fastenings for aluminum do not differ from those for steel or other metals. It is common practice in rivet joint design to base the design upon the tensile, shearing, and bearing strengths of the plates and rivets and to neglect the frictional resistance in the joint. This practice is suggested for aluminum and aluminum alloys. For example, in the case of a single riveted lap joint, the relation between the thickness of the sheet, the rivet diameter and the rivet pitch should be such that the net tensile strength of the sheets will equal the shear strength of the rivets. The relations also should be such that the bearing strength of the sheets is not less than the net tensile strength of the sheets or shear strength of the rivets, a con-

TABLE II

Average Mechanical Properties of Wrought Aluminum Alloys

Alloy & Temper	Ultimate Tensile Strength Lb. per Sq. In.	Yield Point Lb. per Sq. In.	Elongation in 2 Inches, per cent	Shear Strength Lb. per Sq. In.	Endurance Limit Lb. per Sq. In.	Brinell Hardness 500 kg. load, 10 mm. ball
2SO	13000	4000	30	9000	5000	25
2S½H	17000	14000	15	11000	7000	32
2SH	24000	20000	5	13000	8500	40
3SO	16000	5000	30	11000	7000	28
3S½H	21000	19000	15	13000	9500	45
3SH	30000	25000	6	16000	10500	55
17SO	27000	10000	18	17000	11000	45
17ST	58000	35000	20	35000	15000	98
25SO	27000	11000	18	15000	8000	45
25SW	48000	25000	18	30000	14500	80
25ST	58000	35000	20	35000	15000	98
51SO	16000	6000	25	10000	6000	30
51SW	35000	20000	22	24000	10500	65
51ST	48000	35000	14	30000	10500	95

NOTE: "H," in alloy number, denotes hard temper; "O," annealed; "W," quenched; and "T" fully heat treated.

dition which will not exist except in joints having rivets that are unusually large with respect to the thickness of the sheets.

Owing to the unknown amount of cold working that a rivet receives in the heading up operation, there may be some question in regard to the correct shear strength value to use in the design of a riveted joint. Unless specific data on shear strength are available, it would seem reasonable in design practice to base the design of a riveted joint on an assumed value for the ratio between shear strength of the rivet and the tensile strength of the sheet. A value of 0.6 is suggested for this ratio when the sheet and rivet are of the same alloy. The use of this ratio will, in nearly every case, result in a good design. Obviously, in a properly balanced joint the greater the rivet diameter for a given thickness of sheet, the greater will be the pitch of the rivets and the greater will be the computed efficiency of the riveted joint.

TABLE III

Factors Applying to Design Stresses of Wrought Aluminum Alloys at Elevated Temperatures

	Temperature in Degrees Fahrenheit				
Alloy	100	200	300	400	500
2SO	1.00	.80	.60	.40	.30
2S½H	1.00	.60	.45	.30	.20
2SH	1.00	.50	.35	.25	.15
3SO	1.00	.80	.70	.55	.40
3S½H	1.00	.65	.50	.40	.30
3SH	1.00	.55	.45	.35	.25
17SO	1.00	.90	.85	.60	.35
17ST	1.00	.90	.75	.30	.15
25SO	1.00	.90	.70	.40	.35
25SW	1.00	.75	.75	.25	.15
25ST	1.00	.90	.75	.20	.10
51SO	1.00	.90	.65	.40	.30
51SW	1.00	.85	.85	.20	.15
51ST	1.00	.90	.75	.15	.10

While single riveted joints are satisfactory for many classes of work, they are not as efficient as multiple row riveted joints. As in the case of steel and other metals, the latter should be used when higher joint efficiencies are desired. The marginal distance, or distance from the edge of the sheet to the center line of the nearest row of rivets, should not be less than about one and one-half times the rivet diameter for aluminum riveted joints. This distance should not be much greater where caulking is employed to make a tight seam, thus conforming with steel practice. Where caulking is not employed, a greater marginal distance frequently is used to decrease the tendency for the edge of the sheet to become wavy. In multiple riveted joints for aluminum, the transverse pitch or distance between rivet rows should preferably be a little greater than that commonly used in steel practice. As a general rule to fit most cases, it is suggested that the transverse pitch is about 75 per cent of the rivet pitch.

Aluminum screws and bolts usually are made from the high strength aluminum alloys. Alloy 17ST is perhaps the most commonly used although screws and bolts in 25ST and 51ST alloys sometimes are employed. Since the ultimate tensile and compressive strengths of 17ST are approximately 60,000 pounds per square inch and the yield point 35,000 pounds per square inch, it follows that bolts made of this alloy have approximately the same strength in tension and compression as ordinary mild steel bolts. It likewise is true that such bolts will have practically the same strength as steel bolts in flexure or bending.

Compensates for Difference in Strength

The nominal shear strength of this aluminum alloy is approximately 34,000 pounds per square inch and bolts made of it will be about 75 per cent as strong in shear as steel bolts, since the latter have a shear strength of about 45,000 pounds per square inch. This difference in shear strength could be compensated for to a certain extent by increasing the thickness of the bolt head and nut sufficiently to take care of the difference in the strength of the two metals. Bolts of 17ST will, however, be weaker against torsional strains owing to the difference in shear strength, and will therefore be more apt to twist off if drawn up too tightly.

Fig. 7—Dump truck body, and lifting arms and sub-frame of hoisting mechanism are made of aluminum alloys

In view of these conditions, in most cases it is is necessary to use a 17ST bolt about one size larger than would be used in steel, especially when the bolts are to be used in places where they must be drawn up tightly.

General Considerations in Designing Mechanical Springs

By A. M. Wahl

Part II—Stresses Employed in Practice.

NEW spring design formulas based on the results of recent research are among the valuable data presented in this, the second article of the series on spring design by A. M. Wahl, research laboratory, Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

IT IS a fact that the fundamental torsional action present in helical springs has not been well understood until recently. The effect of spring index, i.e. the ratio of mean coil diameter to wire diameter in raising the maximum stress occurring in helical springs has not been quantitatively known with any degree of certainty until recent years.

The object of the author in the accompanying article is to present the latest results of research in a form convenient for the designer, and also to give him some specific ideas and hints which may be used in practical spring design. It also is intended to give some data showing why many of our present formulas for helical springs of round wire are inaccurate, and to show why small variations in spring dimensions often cause large variations in the load deflection rates for the spring. Some comparison of working stresses as used in practice with the tensile and endurance properties of the spring material also will be made according to the method proposed by the author in the first article of this series which appeared in the May issue of MACHINE DESIGN, commencing on page 26.

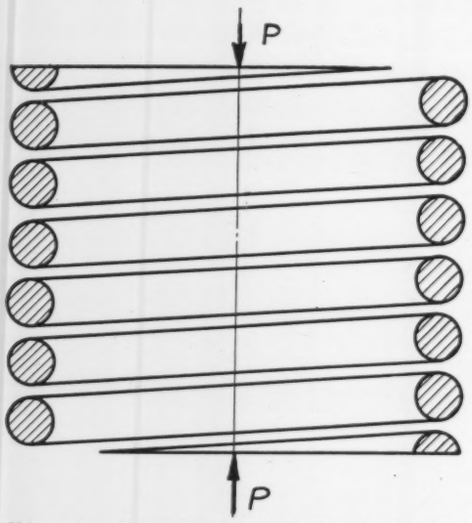


Fig. 1—Round wire helical spring of large index, axially loaded

Helical Spring Formulas

Since helical springs of round wire are of paramount importance in practical design, we first will take up the case of springs loaded axially by a load P as shown in Fig. 1. Commonly used formulas for calculating such springs are based on the torsion of a round bar. If we consider the spring as a round bar under a torsional moment Pr, it can be shown from fundamental formulas that:

$$\text{Maximum stress } S = \frac{16\ Pr}{\pi\ d^3} \quad \ldots\ldots (1)$$

$$\text{Total deflection } \delta = \frac{64 Pr^3 n}{Gd^4} \quad \ldots\ldots (2)$$

where

P=total load on spring in pounds.
r=mean radius of coils = ½ mean coil diameter in inches.
d=wire diameter in inches.
G=shear modulus of rigidity in pounds per square inch.
n=number of active coils.
δ=total deflection of spring in inches.
S=maximum shearing stress in the spring, pounds per square inch, at the given load P.

Recent tests made under the direction of the author have shown that while equation (2) is nearly correct for the deflection of helical springs of round wire, equation (1), which gives the value of maximum stress, is inaccurate for the smaller values of spring index used in practice. For example, the values of stress given in the com-

monly used spring tables are as much as 60 per cent in error for springs having an index of three such as the spring shown in Fig. 2, which has an index about as small as any used in practice.* The reason for this will be explained later. In design, instead of using equation (1), we use therefore the following equation

$$S = K \frac{16\ Pr}{\pi\ d^3} \qquad \text{.....(3)}$$

This equation is exactly the same as equation (1) except that the second member of that equation is multiplied by a factor K which depends on the spring index $2r/d$ and which may be taken from the curve of Fig. 3. The algebraic expression for K is

$$K = \frac{4c-1}{4c-4} + \frac{.615}{c} \qquad \text{.....(4)}$$

In this $c = 2r/d$ = spring index.

It may be seen from this curve that at large values of spring index the value of K tends to approach unity. This means that in such cases the expression for calculating stress given by the ordinary formula, equation (1) is nearly correct. This is what we would expect since at large indices, the spring more nearly approximates a round bar under torsion.

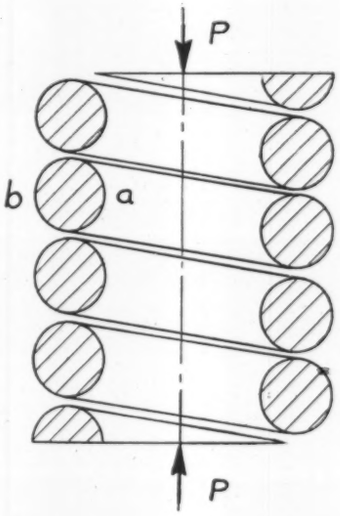


Fig. 2 — Helical spring of small index, axially loaded. The index, or ratio of mean coil diameter to wire diameter, of this spring is about the minimum employed for normal conditions

In using ordinary spring tables, the true stress for any given load or deflection for a given spring may be found if the stress on which the particular table is based is multiplied by the factor K taken from the curve of Fig. 3. For example, suppose we have a spring of 3/16-inch mean diameter made of .062 inch wire, the table being based on a stress of 80,000 pounds per square inch computed by the ordinary formula. The load and deflection for this spring as given in the table are 41.19 pounds and .0145 inch per turn respectively. Since the index of the spring is $c = 2r/d = 3/16 \div .062 = 3.02$, and from Fig. 3 the stress multiplication factor $K = 1.58$, the *true* stress at a load of 41.19 pounds and deflection of .0145 inch per turn will be $80{,}000 \times K = 80{,}000 \times 1.58 = 127{,}000$ pounds per square inch. Suppose now we wish to find the true stress at a load of 30 pounds. Since the stress is directly proportional to the load, the true stress will be, at this load, $S = 30/41.19 \times 127{,}000 = 92{,}500$ pounds per square inch. In this manner, it is possible to calculate the true stress in springs in a simple manner by using ordinary spring tables.

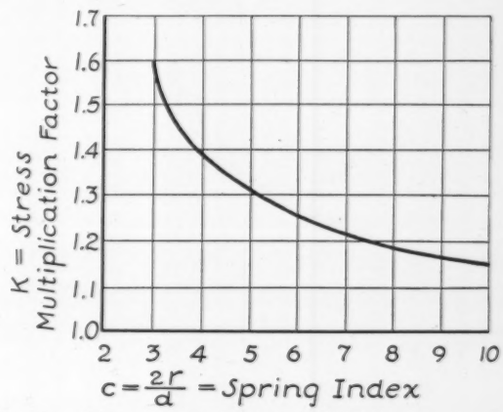


Fig. 3—Curve for determining stress multiplication factor K

Using this example, we know that the true stress at a load of 41.19 pounds is 127,000 pounds per square inch. Now suppose that the designer wishes to stress the spring only to a true stress 80,000 pounds per square inch. The permissible load then is found by multiplying 41.19 by the ratio 80,000/127,000 thus obtaining 26 pounds permissible load on the spring. The deflection for a permissible stress of 80,000 may be found in the same way, since the deflections for practical springs are nearly proportional to loads. In all the above consideration it is, of course, assumed that the material is not stressed above the applicable elastic limit.

Stresses Used in Practice

In looking into the matter of working stresses used in helical springs in practice, the author has found a wide variation in the values of stress employed, if one stops to consider that the ordinary spring tables list springs whose indices vary from 3 to perhaps 12 or more. Nominally these springs have stresses of let us say 80,000 pounds per square inch. However, it will be seen from the curve of Fig. 3 that the actual stress in those springs listed in the table and having indices of three is about $1.6 \times 80{,}000$ or 128,000 pounds per square inch. On the other hand those

*See author's paper, *Transactions*, A.S.M.E. 1929, A P M—51-17.

springs in the table having indices of, say, 12 the true stress will be about 1.1 × 80,000 or 88,000 pounds per square inch. Hence, it will be seen the true stresses employed in the average spring

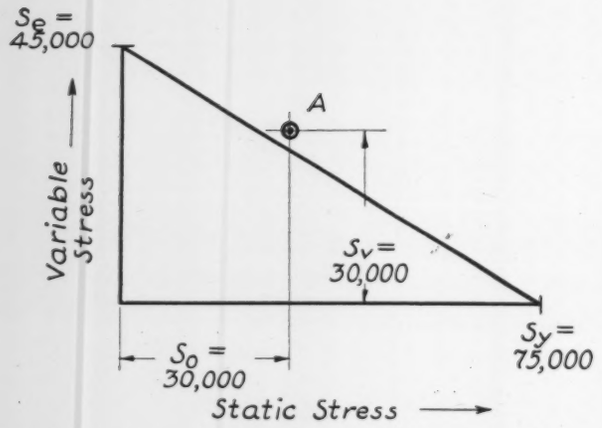


Fig. 4—Assumed endurance diagram showing point A where helical spring works. Stress ranges from 0 to 60,000 pounds per square inch continuously

table may vary widely. The variations in actual practice are even greater.

What values of working stress actually are in use in practice? A common figure given for helical springs is a stress of 60,000 pounds per square inch for severe service. Let us assume that by severe service we mean cases where the spring operates from zero load to a maximum load continuously. We also will assume that the spring index is large so that the true maximum stress in the spring will be about 60,000 pounds per square inch in shear if designed by ordinary spring tables. Under these conditions the static stress will be

$$S_o = \frac{60000}{2} = 30000 \text{ pounds per square inch}$$

and the variable stress will be

$$S_v = \frac{60000}{2} = 30000 \text{ pounds per square inch}$$

We now assume that the spring material is 1 per cent carbon steel heat treated. Fatigue tests on this material at the Westinghouse research laboratories show an endurance limit in reversed flexure of around 90,000 pounds per square inch, and tensile tests show a yield point of 150,000 pounds per square inch determined as the point where the plastic strain is .2 per cent. On the basis of the maximum shear theory the endurance limit in reversed torsion* is S_e = 45,000 pounds per square inch and the yield point in torsion is S_y = 75,000 pounds. We then may plot the endurance diagram as in Fig. 4, the method being the same as that explained in the author's previous article in MACHINE DESIGN. The point where the ordinary helical spring works if S_o = 30,000 and S_v = 30,000 as determined for severe service is shown by point *A* on the figure. From this it appears that the spring if working under such conditions is working close to the endurance limit and has little if any margin of safety. As mentioned in the previous article this endurance curve would be conservative if the actual spring were free from surface imperfections.

*Tests made at the University of Illinois under the direction of Professor Moore indicate that the endurance limit on reversed torsion is about half that in reversed flexure for spring steel. See *The Fatigue of Metals* by Moore and Kommers.

Surface Marks Affect Endurance

But this method does not take into account the reduction in endurance limit caused by surface defects in actual springs and for this reason, it is possible that the endurance curve is not conservative. It therefore is the author's opinion that this figure of 60,000 pounds per square inch for severe service as defined above is high, and that where the consequences of failure are serious, a lower value should be used when the service is such that the spring is stressed continuously from zero to a maximum value.

Let us now assume that by severe service we mean a stress range from say half the maximum stress to the maximum stress continuously. Taking again a maximum stress of 60,000 pounds per square inch, this means a stress range from 30,000 pounds per square inch to 60,000 pounds per square inch, continuously. Under these conditions, the static stress will be S_o = 45,000 pounds per square inch and the variable stress S_v = 15,000 pounds per square inch. Under these conditions, the spring will operate at point *B*, Fig. 5. It may be seen that even under these conditions the spring is operating near the limiting curve defined as above and hence the factor of safety is not great.

This fact is recognized by the writers of handbooks who in certain cases state that for severe applications the stress should be greatly reduced.

Spring Fails Under Service Conditions

The case of an actual spring which failed in service is represented in Fig. 6. This spring was operating under the conditions represented by point *C*, computed by equation (3); namely, a static stress of 53,000 pounds per square inch and a variable stress of 16,000 pounds per square inch. The spring was designed evidently by ordinary formulas, and since it had an index about 3, the true stresses were about 60 per cent greater than those calculated. The point computed by the ordinary helical spring stress formula is

shown by point D, Fig. 6. It may be noted that the spring appeared perfectly safe when computed by ordinary formulas (point D) but was actually above the danger point when computed by the more exact formulas (equation 3). This illustrates vividly the fact that it is more logical to design springs by means of exact formulas which will reveal the true stresses rather than by ordinary formulas.

If designers could design their springs by accurate formulas, and calculate the actual factors of safety by the equation given in the previous article, namely,

$$n = \frac{1}{\frac{S_v}{S_y} + \frac{S_r}{S_e}} \quad \text{.....(4a)}$$

it would be possible to correlate and classify the factors of safety used for various spring applications. In view of the lack of fatigue data on actual springs loaded as in service, such a classification undoubtedly would be of value.

In considering the effect of even small variations in coil and wire diameter on the deflection of helical springs, it is surprising to note that a 1 per cent cumulative variation in coil and wire diameter may amount to a 7 per cent change in the load deflection characteristics of the spring. This may be shown as follows. The deflection formula for helical springs as given by equation 2 is

$$\delta = \frac{64\ Pr^3n}{Gd^4}$$

In this formula r and d are the *nominal* mean coil radius and wire diameter respectively. Suppose now that the *true* mean coil radius and wire diameter are $r_a = r\ (1 + \epsilon)$ and $d_a = d\ (1 + \lambda)$ where ϵ and λ are small quantities. We will then find that the true deflection will be

$$\delta_1 = \frac{64\ Pr_a^3 n}{Gd_a^4} = \frac{64\ Pr^3\ (1+\epsilon)^3 n}{Gd^4\ (1+\lambda)^4} \quad \text{.....(5)}$$

Since ϵ and λ are small relative to unity we may put $(1 + \epsilon)^3 = 1 + 3\epsilon$ and $(1 + \lambda)^4 = 1 + 4\lambda$ with sufficient accuracy, thus neglecting small quantities of higher order. We then obtain from equation (5) the *true* deflection

$$\delta_1 = \frac{64\ Pr^3 n\ (1+3\epsilon)}{Gd^4\ (1+4\lambda)} \quad \text{.....(6)}$$

Since λ is small, 4λ also will be small and we may write

$$\frac{1}{1+4\lambda} = 1 - 4\lambda$$

Then equation (6) becomes

$$\delta_1 = \frac{64\ Pr^3 n\ (1+3\epsilon)\ (1-4\lambda)}{Gd^4} = \frac{64\ Pr^3 n\ (1+3\epsilon-4\lambda-12\lambda\epsilon)}{Gd^4}$$

In this we can neglect the term $12\lambda\epsilon$ with respect to the other terms since it is a quantity of higher order. We then obtain the true deflection

$$\delta_1 = \frac{64\ Pr^3 n}{Gd^4}\ (1 + 3\epsilon - 4\lambda) \quad \text{.....(7)}$$

It is seen that the true deflection is merely the nominal deflection $64\ Pr^3n/Gd^4$ multiplied by a term $1 + 3\epsilon - 4\lambda$ which depends on ϵ and λ. Suppose now that the actual mean coil diameter or radius was 1 per cent greater than the nominal, i.e. $r_a = r\ (1.01)$ or $\epsilon = .01$, while at the same time that the true wire diameter is 1 per cent *less* than the nominal i.e. $d_a = d\ (1 - .01)$ or $\lambda = -.01$. Putting these values of ϵ and λ in (7) we get

$$\delta_1 = \frac{64\ Pr^3 n}{Gd^4}\ (1 + .03 - 4(-.01) = \frac{64\ Pr^3 n}{Gd^4}\ (1.07)$$

In other words under such conditions with a 1 per cent cumulative variation in coil and wire diameter from the nominal values the true deflection will be 1.07 times the nominal deflection or 7 per cent greater.

As a practical example let us take a spring recently examined by the author. This spring was made of 9/16 inch wire or $d = .5625$ inch. After cutting up the spring and measuring the dimensions, it was found that the average wire diameter was .551 inch which would correspond to an error in the wire size of 2.04 per cent. In other words the true wire size varied 2.04 per cent from the nominal. Therefore $\lambda = -.02$. Assuming that the true mean coil diameter of this spring were equal to the nominal, i.e. that $\epsilon = 0$, then from equation (7), the true deflection δ would be $1 - 4\lambda = 1.08$ times the calculated for $\lambda = -.02$. Hence, this spring would have a deflection 8 per cent greater than that calculated if the true coil diameter were equal to

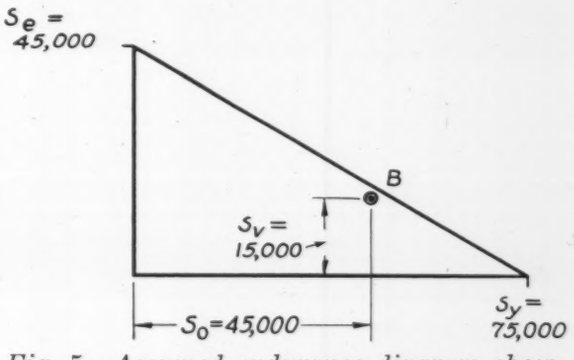


Fig. 5—Assumed endurance diagram showing point B where helical spring works. Stress ranges from 30,000 pounds per square inch to 60,000 pounds per square inch continuously

the nominal. Actually in this spring the coil diameter had been made about 2 per cent less than the nominal which meant that ϵ was —.02. Then from equation (7) the true deflection was $(1 + 3\epsilon - 4\lambda) = (1 + 3\ (-.02) - 4\ (-.02) =$ 1.02 times the nominal deflection. In other words, the spring manufacturer had made the true coil diameter slightly smaller than the nominal value in order to compensate for the decreased diameter of the wire from which the spring is made. In this way he obtained a load deflection curve closely approximating the calculated value, even though the wire was 2 per cent too small.

Since commercial wire stock may vary from the nominal size by even more than 2 per cent, which corresponds to a change in the load deflection characteristic of the spring of 8 per cent, it is clear that if springs are to be held to de-

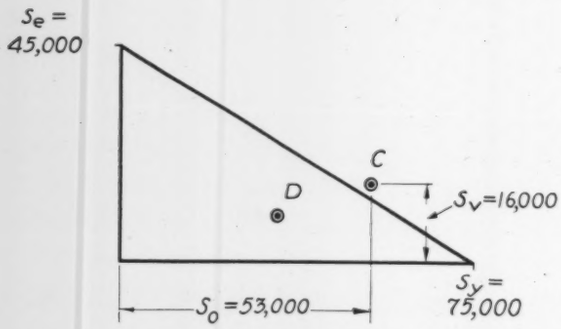


Fig. 6—Endurance diagram for broken helical spring. C=point where failed spring was actually operating. D=point computed by ordinary helical spring stress formula

flections within less than 10 per cent of the calculated value, and are not to be too expensive, it may be necessary to allow the spring manufacturer to compensate for this by changing the coil diameter slightly. Such examples show the necessity of close co-operation between the spring designer and spring manufacturer.

Effect of Dimensions on Stress

It also should be noted that there is a similar effect on stress by changing the spring dimensions. In order to estimate this effect we use the ordinary formula equation (1) which is

$$S = K \frac{16\ Pr}{\pi d^3}$$

where r and d are nominal dimensions as above. Since the effect on K of a slight change in the spring dimensions will be small, we assume that K is constant. If r_0 and d_0 are the *true* dimensions, we have as above, the true stress

$$S_1 = K \frac{16\ Pr_0}{\pi d_0^3} = K \frac{16\ Pr\ (1+\epsilon)}{\pi d^3\ (1+\lambda)^3}$$

Proceeding in the same way as before by neglecting small quantities of higher order we find the true stress equal to

$$S_1 = K \frac{16\ Pr\ (1+\epsilon)}{\pi d^3\ (1+3\lambda)}$$

or

$$S_1 = K \frac{16\ Pr}{\pi d^3} (1 + \epsilon - 3\lambda) \quad \text{.....(8)}$$

Let us take as an example the 9/16 inch wire spring considered above. Since in this case the true wire diameter d_0 was 2 per cent under size i.e. λ = —.02, this will correspond to an increase of stress from (8) of 3λ or 6 per cent provided the coil diameter were unchanged.

Effect of Modulus of Rigidity

Since the deflection of helical springs is inversely proportional to the modulus of rigidity G of the material in shear, it is necessary to use correct values for G if a correct value of deflection is to be had.

Many tests on carbon spring steels made at the Westinghouse research laboratories show a modulus of elasticity in tension of very nearly 30 x 10^6 pounds per square inch in all cases. In the case of some alloy spring steels the modulus was found to be as low as 27.5 x 10^6 pounds per square inch. The modulus of elasticity E bears a definite relation to the modulus of rigidity G according to the theory of elasticity. The relation is $G = E/2(1+\gamma)$ where γ = Poisson's ratio. Taking Poisson's ratio = .3 we have $G = E/2(1 + .3) = E/2.6$. For the case of the carbon steels with a modulus of elasticity E = 30 x 10^6 pounds per square inch the modulus of rigidity G would be 11.52 x 10^6 pounds per square inch. This value of rigidity is confirmed by tests on several carbon steel springs made under the direction of the author.

In the latter the modulus of rigidity was determined by measuring deflections in the body of springs having large indices and later cutting up the springs so that the wire diameter could be measured accurately at all points. Knowing the spring dimensions, G could be computed from equation (2). Three carbon steel springs of about 1 per cent carbon, heat treated from one manufacturer, tested as follows. Average G = 11.47 $\times$ 10^6 pounds per square inch; maximum variation from average 0.26 per cent. Three carbon steel springs oil tempered of about 1 per cent carbon steel, from a different manufacturer tested as follows. Average G = 11.23 $\times$ 10^6 pounds per square inch; maximum variation from average = 0.63 per cent. These tests indicate that the actual modulus of rigidity of the 1 per cent carbon steel springs is not far from the value predicted from the known value of the modulus of elasticity in tension.

Employing Antifriction Bearings in Heavy Machinery

By John D. Rauch

A BACKGROUND of many years experience in the design and construction of cranes, hoists, power shovels, conveyors and locomotives renders Mr. Rauch, now chief engineer of the Ohio Power Shovel Co., Lima, O., particularly capable of preparing the accompanying article from an authoritative angle.

MACHINERY for hoisting, elevating and conveying materials represents one of the oldest and most extensive applications wherein a combination of gears, shafts, clutches, drums, and in most cases sheaves, are employed for transmission of power. The number of combinations that have been worked out employing the above component parts almost is beyond conception, each design striving for betterment in general arrangement and simplicity in assembly.

From time to time revolutionary features have been injected into the designs which have tended to increase the efficiency and life, such as the use of steel castings instead of iron, and the introduction some twenty years ago of machine cut gears, both of which now have become almost universal.

This process of refinement with a view of increasing efficiency through the adoption of new materials and the application of new features is going on indefinitely. Problems confronting a designer of such machinery today not are only confined to simplifying the general design, but to the application of modern practices and advanced ideas through intensive research work.

Opportunity Lies in Development

One of the greatest opportunities along this line lies in the development of antifriction bearings and in their use under conditions for which plain bearings might be considered unsuitable. This is a subject of paramount importance, yet at the present time, outside the automotive field, comparatively few really successful examples can be cited. Mere substitution of such bearings for bronze or babbitt bearings in existing designs generally is not practical from either a constructive or assembly viewpoint. In other words the machine must be designed around the bearings rather than the bearings designed for the machine.

It only has been in the last few years that antifriction bearings of either the roller or ball types have been perfected in sizes and capacities for successfully coping with heavy shock loads, therefore a discussion of their use and application seems particularly pertinent at this time. Certain charteristics of antifriction bearings make them desirable and applicable to this type machinery.

Antifriction Bearings Have Important Uses

Economies resulting from the use of this type of bearing are much farther reaching than imagined at first thought. To the layman the use of antifriction bearings appeals only from the standpoint of quietness, elimination of friction and therefore less power consumption, but to the designer experienced in the effect of such bearings in heavy machinery their use means much more.

For instance, we will consider two shafts which

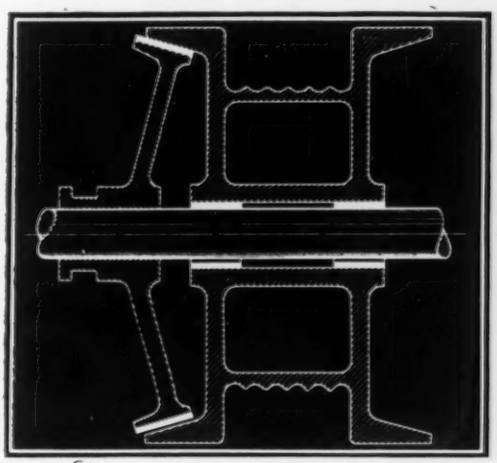


Fig. 1—Eccentricity of cone clutch and cable drum as shown above, causes excessive jerking and grabbing action when clutch is applied

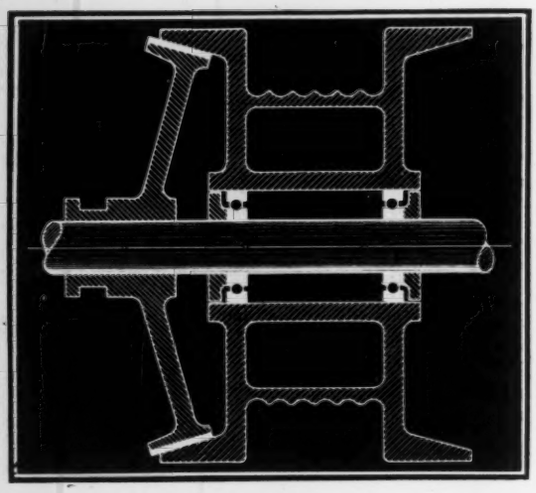


Fig. 2—Application of antifriction bearings though more costly, eliminated eccentricity and provided smooth drive.

revolve through the medium of spur gears. If these gears are accurately cut and the shafts can be held indefinitely at the correct center to center distance, the destructive wear on the gears practically will be nil. If, however, the shafts cannot be held indefinitely at their correct center distance and are allowed to separate even the slightest amount, the wear on the gear teeth is multiplied many fold and in time involves heavy renewal or maintenance costs, and stoppages in production.

This may be the case, on heavy machinery, when shafts are supported in bronze or babbitt bearings. No matter how much care and attention is given to the bearings it is a fact that they wear, and in wearing the center distance between shafts is increased. This effect is eliminated where antifriction bearings are used and properly applied. Where the correct size and capacity of such bearings are employed, it hardly is conceivable that in the life of the machine enough wear could take place in the bearings to change the center distance of shafts enough to cause improper meshing with consequent wear on the gears. In fact this has been demonstrated in actual practice in many cases recently.

Extra Cost Is Justified

This in itself justifies the use of antifriction bearings and more than makes up in savings, the extra cost of their application over the bearings more commonly used.

In the class of machinery under discussion, the use of clutches is common practice and the smooth and successful operation of such clutches, whether they be outside contracting, inside expanding or cone, depends on the driving and driven members always being concentric. As an example, we will take the subject of a cable drum mounted loosely on the shaft and to which the power is applied through the medium of a friction clutch. Usually such a drum is placed on the shaft between gears or other rigid members and it, therefore, is more or less a difficult job to remove the drum in order to renew the bushings. For this reason the bushings are allowed to run long after excessive wear has taken place. This excessive wear in drum bushings always is accompanied by a jerking, grabbing action when the clutch is applied, due to the fact that the wear in the bushings allows the drum to drop eccentric to the cluch, as illustrated in Fig. 1. Before the clutch can function properly the drum has to be bounced around to a central or concentric position. This causes jerking or grabbing with attendant excessive wear on the clutch lining and destructive vibration to the entire machine.

Assures Smooth Operation of Clutch

The use of antifriction bearings, either roller or ball in the drum, as shown in Fig. 2, completely eliminates this condition. In the life of the machine hardly sufficient wear could take place on the bearings to cause eccentricity enough to affect the operation of the clutch and no matter what type of clutch is employed the operation is, therefore, smooth and exertionless. The saving in power and wear and tear of the machine parts more than offsets the additional cost of the application of the antifriction bearings.

In the examples outlined above, the loadings have been considered as purely radial. Where

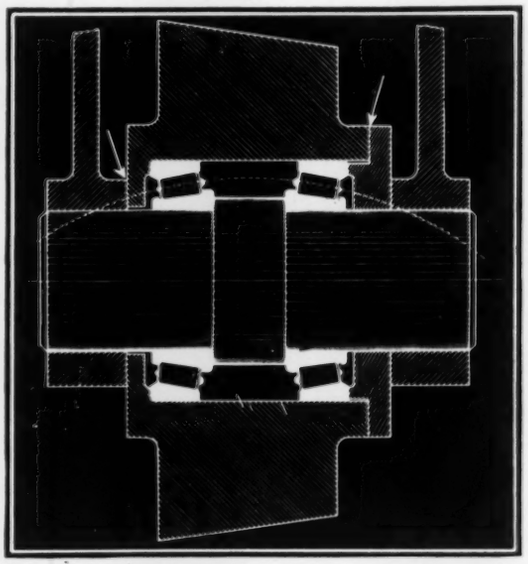


Fig. 3—Note tendency for unit to roll on dotted line when bearings are disposed as indicated in the above illustration

a thrust load is encountered the advantages accruing from the use of antifriction bearings are even more marked. A good example of such an application is a turntable roller. Usually there are a number of such rollers employed for the support of the turntable which revolves around a center pin. The load is not always equally disposed on these rollers and where they are equipped with bronze or babbitt bearings and bronze thrust washers, greater wear will be encountered on some than on others. This condition allows the rollers that are receiving the greatest wear to work outward to a greater radius than the others. The effect of this is to cause an uneven wear on the roller path which in turn causes undue friction, vibration and uneven operation of the machine.

Care Necessary in Application

The application of antifriction bearings designed for both radial and thrust loads under a condition of this nature maintains all of the rollers in perfect alignment indefinitely, with smooth wearing of the roller path and total elimination of vibration. In applying such bearings, however, care is necessary that they be properly disposed. For instance, in Fig. 3, it will be noted that there might be a destructive tendency for the whole unit to roll on the bearings as indicated by the curved line with consequent binding at points shown by arrows, but by applying the bearings as shown in Fig. 4 this tendency is eliminated entirely.

Many other problems wherein standard practices cannot be followed confront the designer. The position of the bearings, as well as the stresses imposed upon them must be considered carefully, as well as the question of assembly. Simple layouts can be made, that is, layouts which look simple on paper, but in which the question of ability to assemble economically makes them impractical. There are certain applications where a ball bearing is the ideal type and would

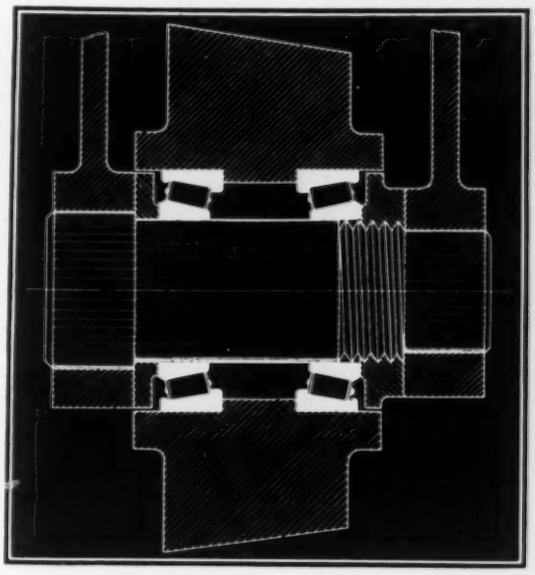


Fig. 4—Slightly more machining and expense are involved with this arrangement, but rolling tendency is eliminated entirely

permit of easy assembly; for other applications the straight roller bearing would prove better, and there are still others where tapered roller bearings would be the more economical.

Too much thought and study cannot be devoted to this timely subject by those interested in the advancement of heavy machinery, and while only a few of the more important problems attending the application have been referred to in the foregoing, others will be met with in the design requiring much skill and patience on the part of the designer.

Combating Vibration in Centrifugal Machinery

(*Concluded from Page* 20)

friction bearings are used throughout and are lubricated by five Alemite fittings which need no attention for months at a time. Automatic brake application is accomplished by a simple mechanical linkage of brake lever and timer, thereby eliminating the need of solenoids and other electrical apparatus which is apt to be more or less complicated.

So at last the extractor operator has been provided with a machine of which he need have no fear; an extractor beside which he can stand while it is operating at a higher speed than was ever formerly practical and still not be aware that it is running unless he looks at the signal light; an extractor on the curb of which a coin can be placed on edge without falling over during the entire running period.

And all because a machine designer remembered that the operator of his product was a human being and deserved the same consideration that he himself would expect. As to whether or not it pays,—well, the great demand for this type of machine is the most substantial form of testimony that it most assuredly does.

User Discusses Merits of Modern Lubricating Systems

Circulating, Forced Feed and Centralized Systems Offer Facilities for Meeting Wide Range of Oiling Problems

PROPER lubricating methods must be applied to machines in order to obtain efficient and uninterrupted operations. In the accompanying digest of a paper, "Steel Plant Lubrication," prepared by W. A. James, the three modern systems are discussed. Mr. James who is chief engineer, Lackawanna plant, Bethlehem Steel Co. presented this paper before the spring meeting of the American Iron and Steel institute at New York recently.

IT TAKES a trained engineer properly to appreciate the terrific toll of friction upon power. The more efficient and more powerful the machine becomes, the greater becomes the friction, and, therefore, the greater the need for highly effective lubrication of the machinery.

One of the most outstanding of modern lubricating methods is the so-called "circulating system," which is becoming the chief method of lubrication for the most up-to-date steel mills. It includes a main supply line, with branches to the various elements to be lubricated, a common return line to take the individual returns from each element, a suitable tank for receiving and cooling the return oil and separating water and heavy impurities from it, and a means of cleaning and purifying the oil, as well as storage for the clean oil.

Solves Many Lubrication Problems

A second comparatively new method of lubrication is what is known as "forced feed" lubrication. This has proved the solution to many problems, particularly those involving individual machines such as punches, drills and others, where operation is intermittent. Systems of this kind are operated at varying pressures from the machines to which they are attached. Forced feed pumps can be constructed with any number of oil feeds, which can be piped to the different bearings. The amount of oil supplied to each bearing can be accurately regulated through adjustment of the stroke of the pump plungers to suit the individual bearing requirements.

A third form of lubrication developed only within the past few years and now coming into rather general application, is the "centralized system," which uses either oil or grease. Manually operated one or more times during the day, it forces the lubricant to the bearings on the machine or equipment. Its application has been particularly successful with respect to individual machines, and it has developed into a useful method of lubricating overhead cranes.

Modern lubricating practice has demonstrated that it is not necessary to develop heat in steel mill roll necks by friction in order to melt the grease, and thus lubricate them.

It will be found that if bearings are properly designed, and the correct lubricant rightly is applied, there should be no occasion for neck-bearings to run warm, which indicates unnecessary and avoidable friction. Automatic lubrication for this part of mill equipment can improve conditions considerably. Experiments with a "forced feed" system for lubricating the necks have already demonstrated that this can effect a reduction in bearing changes in the unit of about 50 per cent.

The bearings of hot and cold saws always have presented a somewhat troublesome problem in lubrication, on account of their high speeds and severe operation. Adoption of antifriction bearings on the saw spindles has in many instances satisfactorily solved the bearing problem. For these bearings 1½ per cent lime soap base grease, compounded with 1 per cent of manufactured graphite, has proved efficient as a lubricant.

Turbines often are provided with a small oil container in their base. Experience has shown that these containers usually are too small, and that it is advantageous to supplement them with a standpipe of oil in series with the system. This means that the oil is not worked so hard and has a chance to cool and settle, while it will be kept in fair condition for several years without centrifuge or filter. If machines are to run well every advantage must be taken of modern lubrication.

Tests Yield Data for Applying Chrome Nickel Steels

Corrosion Resistance of New Alloys Under Varying Conditions is Worthy of Study to Insure Correct Application

SOME characteristics of two of the corrosion resistant steels more commonly employed in the design of machinery, KM1 and KA2, were summarized in an article appearing in the May issue. On the accompanying pages further information will be presented, particularly in connection with corrosion resistant qualities and machineability of the alloys.

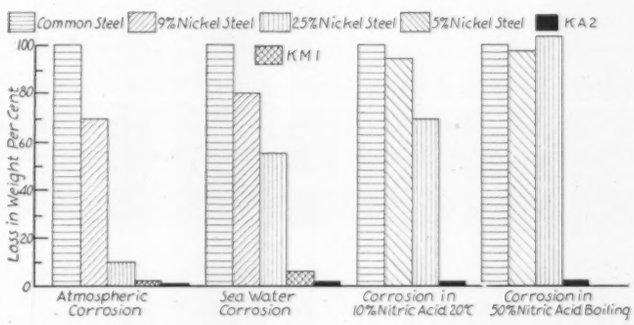


Chart showing comparative examples of resisting power of steels to corrosion

Data which have been compiled to cover corrosion resistance of heat treated and annealed KA2 exposed to dilute solutions, as compared with KM1 and plain carbon steel under the same conditions, show losses in weight in grams per square meter after exposure for 100 hours at 20 degrees Cent. (68 degrees Fahr.) Data are shown in Table I.

Resistance of KA2 and KM1 to air, gases and water, taking common steel on the basis of 100, is shown in Table II and corrosion of KA2 when attacked by commercial chemicals develops as shown in Table III. Comparison with the alloy KM1 is not made in the latter table in view of the fact that it is used for more general design applications where strength is a serious consideration, rather than for parts subject to heavy acid attack.

In connection with the workability and machinability of the two alloys, as mentioned in the previous article it is not possible to deep draw or spin KM1 satisfactorily nor should any attempt be made to weld it. On the other hand, these processes are practical in the case of KA2 when properly heat treated. Both alloys can be forged, but the treatment differs slightly. Forging of KA2 should be carried out as follows:

1. Billets less than 10 sq. in. cross section.

   Heat quickly to about 2200 to 2250 degrees Fahr. Do not soak. Commence forging at this temperature and finish not below 1850 degrees Fahr.

   If forging cannot be completed in one heating, the forging should be returned to the furnace and reheated to the proper temperature.

2. Billets larger than 10 square inches in cross section.

   Preheat and soak billet to 1500 degrees

TABLE I

Losses by Exposure to Dilute Solutions

Solutions	Grams per Square Meter Common Steel	KM1	KA2
Well water	1.00	0.00	0.00
Pure water and CO_2	28.03	0.13	0.03
Magnesium Chloride, 0.5%	5.44	0.23	0.00
Sodium Chloride, 2.5%	5.00	1.02	0.00
Sulphuric acid, 1/10 normal	191.00	.....	0.05
Hydrochloric acid, 1/10 normal	191.00	170.00	0.15
Nitric acid, 1/10 normal	147.50	2.35	0.00
Sulphurous acid, 1.5%	491.00	152.80	0.00
Acetic acid, 1/10 normal	29.60	0.81	0.00
Tartaric acid, 1/10 normal	54.80	8.02	0.00
Tannic acid, 1/10 normal	38.00	0.60	0.00

TABLE II

Losses by Exposure to Air and Water

	Grams per Square Meter Common Steel	KM1	KA2
Air	100	0.40	0.00
Air, heavy with smoke	100	0.93	0.86
Air stream, 1292° Fahr.	100	0.50	0.50
Air stream, 1508° Fahr.	100	1.80	1.50
Well water	100	0.00	0.00
Sea water	100	5.20	0.60
Superheated steam and air at 752 degrees Fahr.	100	3.20	5.90
Mixture of steam, air and sulphurous acid at 752 degrees Fahr.	100	8.10	2.50
Mixture of steam, air and hydrochloric at 752° F.	100	28.40	9.50
Mixture of steam, air and ammonia at 752° Fahr.	100	1.75	0.72

Fahr. maximum. Then raise the heat rapidly to 2200 to 2250 degrees Fahr. Do not soak at the high heat. Then forge as previously mentioned.

(Note: In either case, the first blows of the hammer should be light.)

Treatment in forging KM1 is as follows:

Heat carefully to about 1400 to 1475 degrees Fahr. then rapidly to the forging heat of 2000 to 2100 degrees Fahr. Do not soak. Reheat when heat drops below 1650 degrees Fahr. Cool finished forgings slowly under cover.

TABLE III

Losses by Exposure to Various Chemicals

	Temperature, degrees Cent.	Grams per hour per sq. meter
Nitric acid, 1:10	20	0.00
Nitric acid, concentrated	20	0.00
Nitric acid, 1:1	Boiling	0.04
Nitric acid, concentrated	Boiling	0.02
Nitric acid plus 5% sulphuric acid	Boiling	0.59
Sulphuric acid, 10%	20	0.07
Sulphuric acid, 30%	20	0.16
Sulphuric acid, 66%	20	0.001
Sulphuric acid, 98%	20	0.012
58% sulphuric acid, 40% nitric acid, 2% water	20	0.00
58% sulphuric acid, 40% nitric acid, 2% water	60	0.05
58% sulphuric acid, 40% nitric acid, 2% water	100	0.7
10% sulphuric acid saturated with copper sulphate	20	0.00
Acetic acid, 1:1	Boiling	0.03
Acetic acid, concentrated	Boiling	0.60
Phosphoric acid, 10%	Boiling	0.01
Phosphoric acid, 45%	Boiling	0.04
Sulphurous acid plus 1% sulphuric acid	10	0.05
Boric acid, concentrated	100	0.00
Oxalic acid, concentrated	20	0.00
Oxalic acid, concentrated	40	0.01
Lactic acid	20	0.00
Butyric acid	20	0.00
Butyric acid, saturated	130	0.00
Oleic acid	150	0.03
Gallic acid, saturated	Boiling	0.00
Linseed oil plus 3% sulphuric acid	200	0.05
Carbolic acid, crude	90	0.05
Citric acid, 5%	Boiling	0.00
Sodium chloride, saturated	Boiling	0.10
Sodium chloride, 25%	Boiling	0.03
Sodium chloride, 10%	Boiling	0.00
Sodium sulphide, 1:1	90	0.02
Potassium chlorate, saturated	Boiling	0.00
Potassium hypochlorite	20	0.01
Potassium hypochlorite	105	0.53
Potassium bitartarate, saturated	Boiling	0.09
Ammonium nitrate	107	0.02
Copper nitrate, 1:1	Boiling	0.00
Copper sulphate, 1:1	Boiling	0.00
Zinc chloride degrees beryllium	35	0.04
Stannous chloride	50	0.27
Nitre	20	0.52
Mercuric bichloride 0.7%	20	0.88
Ammoniacal water	Boiling	0.00
Calcium chloride	100	0.00
Ammonium chloride	Boiling	0.00
Sodium hydroxide, 20%	110	0.02
Sodium hydroxide, 34%	100	0.00
Sodium hydroxide	318	0.22
Potassium hydroxide, 27%	Boiling	0.00
Potassium hydroxide, 50%	Boiling	0.40
Chlorine, iodine and bromine and their halogen acids		dissolve

Fabrication of KM1 and KA2 in respect to pickling, machining, grinding and polishing follows similar lines. Therefore the information summarized hereunder applies to both alloys.

EXPANDING, ROLLING, BELLING, FLANGING and BENDING OF TUBES, ETC. Various operations of expanding, rolling, belling, flanging and bending can be done at room temperature. For bending to short radii, however, the material should be heated between 1000 and 1200 degrees Fahr. and in cases where the equipment is not sufficiently strong to handle steel at these low temperatures, the metal should be heated between 1900 and 2200 degrees Fahr. For bending, upsetting or vanstoning, however, the alloys never should be heated in the range from 1200 to 1900 degrees Fahr.

SHEARING: For shearing the shear blades should be adjusted closely so as to prevent dragging of the metal between the knives.

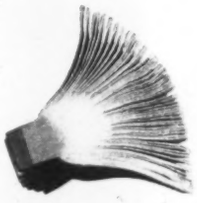
PUNCHING: The materials will not snap off or fracture after a certain portion has been cut as happens in the case of ordinary steel. This metal therefore, must be cut throughout its entire thickness. For this reason, a close fit is required between punch and die which is less than the customary standard clearances used on steel.

DRILLING: Inasmuch as the work hardens under friction, all drills should be sharp and the point of the drill should not be allowed to ride on the metal without cutting. Either a carbon steel drill, ground sharp, or a high-speed steel drill, ground somewhat flatter than standard practice, should be used; in either case, grind off the extreme point of the drill. Apply sufficient pressure to make the drill cut all the time. Laying out should be done preferably with a triangular nosed center punch and the center mark should not be made deeper than necessary. Backing-up should be done in such a way as to permit the drill to cut all the way through without pushing the material away from the drill point. After each hole is made, the drill should be cooled in water. The drill speed with light pressure is similar to that used on mild steel, but the drill should be spotted.

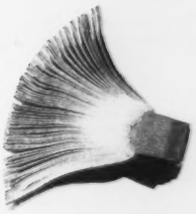
MACHINING: Best results are obtained by the use of slow cutting speeds and moderately heavy cuts. Tools should be ground sharp and kept sharp, with a steep lip and side rake of at least 15 degrees. The materials can be machined dry.

GRINDING AND POLISHING: The amount of grinding and polishing required depends largely on the mill finish of the sheets used. In general a hot-rolled pickled sheet requires Nos. 80, 120, 150, 180, and 200 grinding, whereas sheets and strips which have received some cold rolling, require Nos. 120, 150, 180, and 200 grind or less. It is essential that all marks and scratches from previous grinding be removed before starting with a finer grinding.

Proposes Unconventional Tooth Form for Non-Metallic Gears



Timely Subject Is Discussed at Annual Meeting of American Gear Manufacturers Association



THAT the teeth of non-metallic gears should be of all-addendum form was the recommendation contained in a report by Prof. Earle Buckingham presented at the recent fourteenth annual meeting of the American gear Manufacturers association held at Cincinnati. Prof. Buckingham's proposals are of extreme interest to the design profession and his report therefore is given in full on the accompanying pages of this article.

NON-METALLIC gears should be considered in a class distinctly apart from metallic gears because of the widely different physical characteristics of the materials in comparison with the characteristics of metals.

No material is an exact substitute for any other material. All have their own peculiar and distinct characteristics so that under certain specific conditions, each material has a field of its own where it is superior to any other. Such fields may overlap to some extent, and only in such overlapping fields are different materials directly competitive. Thus, for example, steel is more or less ductile depending upon its physical condition, has a high tensile strength and high modulus of elasticity. Cast iron, on the other hand is not ductile, has a low tensile strength but a high compressive strength, and a low modulus of elasticity. Hence when stiffness and high tensile strength is essential, steel is far superior to cast iron. On the other hand, where these two preceding characteristics are unimportant, but high compressive strength and a moderate amount of elasticity is essential, cast iron is superior to steel.

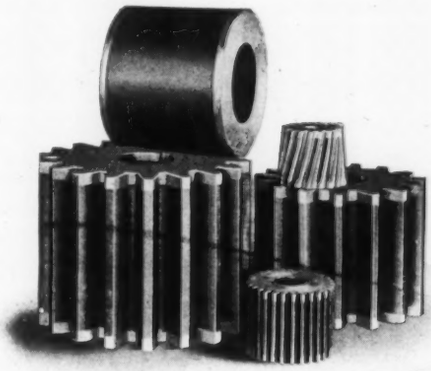
Considering the phenolic laminated materials used for gears, the tensile strength of these materials is slightly less than that of cast iron, the material is far softer than any metal, while its modulus of elasticity is about one-thirtieth that of steel. In other words, if the tooth load on a steel gear which causes it to deform one thousandth of an inch were applied to the tooth of a similar gear made of phenolic laminated material, the tooth of the non-metallic gear would be deformed about one thirty-second of an inch, a striking comparison.

Under these conditions, several things will happen. With all gears, regardless of the theoretical duration of contact, one tooth only will carry the load until this load is sufficient to deform the tooth the amount of the error that may be present. On well made commercial gears of 6DP and finer, the average normal error will be about .002-inch. On very carefully cut gears, this error may be reduced to about one-quarter of this amount.

Elastic Limit May Be Attained or Exceeded

On metal gears, when the tooth has been deformed the amount of the error the stresses set up in the materials may approach or exceed the elastic limit of the material. Hence for standard tooth forms and those generated from standard basic racks, it is dangerous to calculate their strength as much greater than that which can be carried safely on a single tooth.

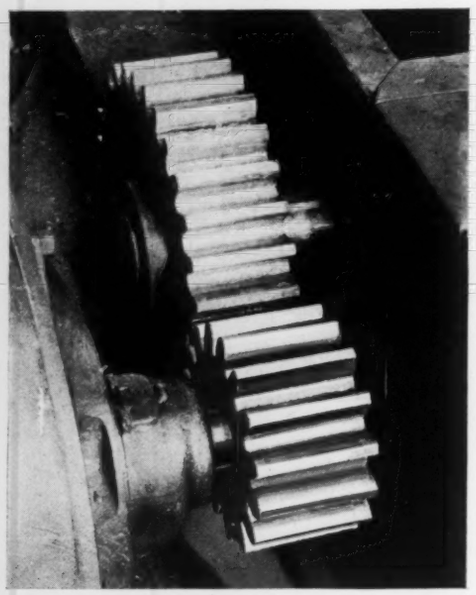
On gears made of phenolic laminated materials, on the other hand, the teeth will deform the amount of this normal error without setting up any appreciable stresses in the material so that the load actually is supported by several teeth. Considering this condition alone, it would be safe to introduce a contact factor



Photograph, courtesy Chicago Rawhide Mfg. Co.

Some types of non-metallic gearing in everyday use

into the Lewis formula for their strength, and this contact factor generally would be larger than the theoretical duration of contact in terms of tooth intervals on the other hand, the teeth will deform the amount of this normal error without setting up any appreciable stresses in the material so that the load actually is supported by several teeth. Considering this condition alone, it would be safe to introduce a contact factor into the Lewis formula for their strength, and this contact factor generally would be larger than the theoretical duration of contact in terms of tooth intervals because the actual point of contact on successive teeth will vary from near the tip to near the root or bottom of the active profiles. Also when the theoretical duration of contact is less than two



Photograph, courtesy Westinghouse Electric & Mfg. Co.

Heavy duty Micarta gear driving from a 50 horsepower motor

teeth, actually three or more teeth may be in simultaneous contact because of the deformation.

But this deformation also introduces another condition. Assuming a non-metallic driving pinion of conventional tooth form, when the deformation becomes appreciable, the corner of the entering tooth of the metal driven gear will make premature contact and start to cut or scrape a free path for itself into the flank of the non-metallic pinion. This condition also is present, but to a much lesser extent, on metal gears when the wear is excessive. The path of the corner of the driven gear as it sweeps into and out of engagement below the pitch line of the pinion is in the form of a loop known as a trochoid. Thus such worn tooth form tend to have a profile of double curvature. The same condition is present when a hob or other generating tool extends so far below the base circle of an involute that undercutting takes place. As a matter of fact, the form of the fillet on any generated gear, whether undercutting takes place or not, is primarily of this same trochoidal form. Hence the additional contact of non-metallic gears is of no practical value if it must be accompanied with this undercutting because of the deformation and scraping action of the entering tooth of the metal gear. This not only will cut away part of the active profile but also will weaken the tooth, thus permitting additional deformation, additional cutting and eventually causing failure.

For non-metallic driving pinions it is possible to minimize this condition of cutting but only by departing from conventional tooth proportions. In other words, in order to obtain the full benefits of the peculiar physical properties of this phenolic laminated material, it is necessary to design the gear teeth themselves to favor the behavior of the material and to minimize the detrimental influence of the relatively large amount of deformation. This can be done by making the tooth of the non-metallic driving pinion all addendum while that of the metal driven gear is made all dedendum. In other words, the pitch circle of the non-metallic driving pinion should be at the bottom of the working depth while that of the metal driven gear is at its outside diameter. In this case, the path of the entering corner of the metal driven gear in relation to the non-metallic driving pinion will be in the form of a cycloid, a form of path which will have but little tendency to cut away the tooth of the pinion.

Standard Basic Rack May Be Utilized

Such a precedure does not require the introduction of any new form of basic rack; any of the present standard ones may be used. For many reasons, the 20 degrees stub tooth form would appear to be the logical selection. This form would give the smallest all-addendum pinion and smallest all-dedendum gear with favorable tooth shapes. It also is the strongest form of the standard ones and thus would favor the low tensile strength of this material.

To summarize; for replacement gears where standard tooth forms and proportions must be maintained no contact factor is used in the Lewis formula when applied to the calculation of the safe load of phenolic laminated gears. For driving pinions of this material of all-addendum form, contact factors varying from about 1.5 up to 4.0 may be used safely. In other words, by modifying the design of such driving pinions, to favor the peculiar physical properties of these materials, several times the load that is safe on pinions of conventional design may be transmitted safely.

PROFESSIONAL VIEWPOINTS

Publication of letters does not necessarily imply that MACHINE DESIGN *supports the views expressed*

Comments from Our Readers. Machine Design Will Pay for Letters Suitable for Publication

Should It Be Draftsman or Ingenieur?

SINCE the first letter appeared on this subject many comments in addition to those already published, have been received from readers. Opinions expressed have varied and it would be difficult to reach a definite conclusion. In view of this and the fact that there does not appear to be any useful purpose gained by publishing these later letters in their entirety, it is proposed to terminate the discussion with a brief summary of some of the views expressed.

One reader says "Old Draftsman forgets or never perceived, that the conditions he describes are in reality only a camouflage to procure for many industrial plants and corporations the services of scientifically trained designers at a lower compensation." Another heartily agrees with J. F.'s letter concerning the misfit name of "Draftsman," and still another suggests that the title "Engineer" in the mechanical fields be reserved for men holding the degree of mechanical engineer.

—THE EDITORS

Calculating Stresses in Springs

To the Editor:

A LAYOUT of a slide rule for calculating the deflection in inch pounds per degree of helical springs in torsion is shown in the accompanying drawing. It is realized, of course, that a slide rule is not accurate enough for some purposes, but this rule will be found convenient in obtaining a base to work on. It is not always an easy matter to arrive at the dimensions of a helical spring in torsion without some tool of this nature to work with.

In addition to indicating points on the rule for brass, phosphor, bronze, monel metal and steel, I have set forth the modulus of elasticity for 10, 15 and 20 million.

Scale 1 and 4 are stationary and have a fixed relation to each other as shown. Scales 2 and 3 are movable, and can be made up and mounted to suit the individual.

As an actual example, let us take a steel spring of the torsional type having .148-inch wire diameter, 2-inch mean diameter, 7½ active coils.

To the mean diameter of 2-inch on scale 1, set the indicator of scale 2. Opposite the wire size of .148 on scale 2, set indicator of scale 3. Opposite the number of coils 7½ on scale 3, read the answer 0.23 inch pounds per degree, on scale 4. Actual test of this spring gave a result to 0.224 inch pounds per degree.

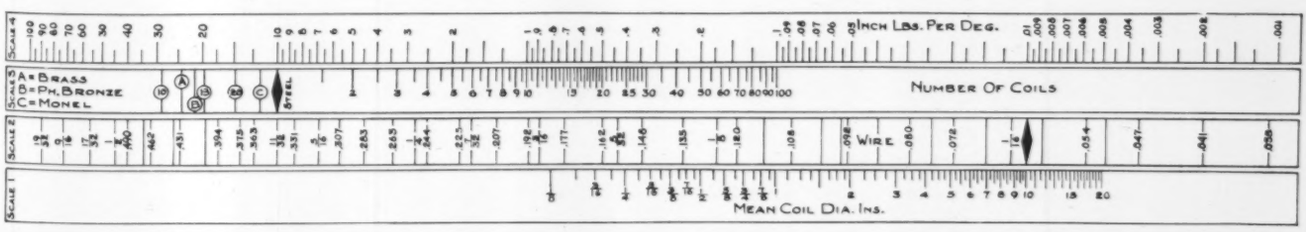
In order to facilitate accurate readings, the drawing could be enlarged and the enlargement pasted on a wood slide rule.

—C. H. THOMPSON,
New York.

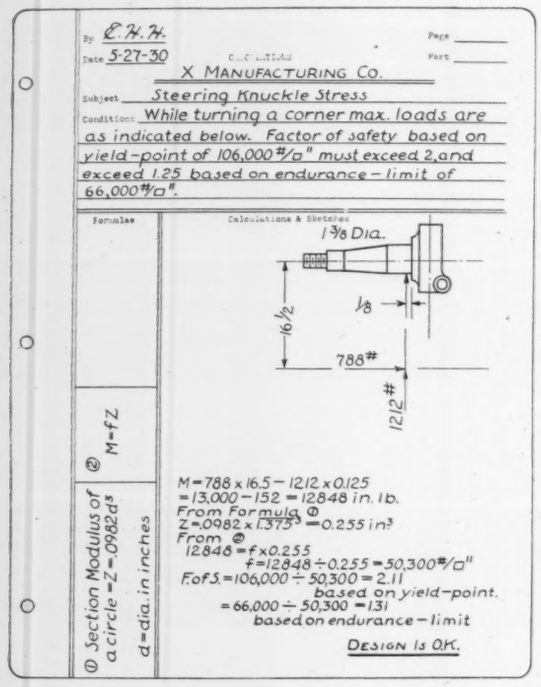
Suggests Filing Design Calculations

To the Editor:

COMPUTATION is the basis for all modern machine design. It is as much a part of any given design as is the part drawing and



should be as carefully preserved. Most modern engineering offices require that all calculations be preserved on a prescribed form which may range from a plain sheet of paper of a given size to a printed form. Reference to past work makes


By E.H.H.
Date 5-27-30
Page
Part
X MANUFACTURING CO.
Subject Steering Knuckle Stress
Conditions While turning a corner max. loads are as indicated below. Factor of safety based on yield-point of 106,000 #/□" must exceed 2, and exceed 1.25 based on endurance-limit of 66,000 #/□".
Formulas
Calculations & Sketches
1 3/8 Dia.
1/8
16 1/2
788#
1212#
② M=fZ
① Section Modulus of a circle = Z = .0982 d³
d = dia. in inches
M = 788 x 16.5 − 1212 x 0.125
= 13,000 − 152 = 12848 in. lb.
From Formula ①
Z = .0982 x 1.375³ = 0.255 in³
From ②
12848 = f x 0.255
f = 12848 ÷ 0.255 = 50,300 #/□"
F of S. = 106,000 ÷ 50,300 = 2.11
based on yield-point.
= 66,000 ÷ 50,300 = 1.31
based on endurance-limit
DESIGN IS O.K.

possible consistent design and in case of error responsibility is located and dispute eliminated.

Compare this with an office where no regulations are in effect covering the preservation of this phase of design. Here the work is jotted on scraps of paper which soon are lost. Even if they are not, the work on them usually is meaningless in a few days, even to the author himself, because of the haphazard, disconnected and incomplete form in which they were written.

Often calculations are not made at all since it is easier to guess at a result. In those offices where it is not required to perserve calculations each designer should keep his own book. Such a book induces the habit of accuracy and soon becomes a valuable reference book.

All the data pertaining to a given problem should be noted down in a clear and intelligible manner, illustrated with a sketch if possible. To guarantee that this is done the writer uses the form illustrated, which is 8½ x 11 inches in size and printed by a mimeograph machine on a fair grade of bond paper. The illustration perhaps is self explanatory but a few words of description may not be amiss.

All steps of a calculation are indicated, but such routine work as addition of figures and logarithmic work is done on the back of the sheet. In this manner all work is preserved and the main body of the work does not become cluttered in appearance. The form is large enough for most calculations, especially if the problem is broken up into distinct parts. Suppose, for example, it is desired to determine the transverse strength of a beam with a complicated section. This can be divided into two steps. First, on one sheet, the section modulus is computed. Second, on another sheet, the actual load conditions of the beam are indicated and its strength determined. In any case, if the problem requires it, the first sheet can be followed by a second one, perhaps a plain sheet. The final results can be underlined in red. Large, sketches can be placed on a separate sheet, folded if necessary and inserted in their proper places.

Indexing the calculation book is worth the time required.

Finally, no matter what form is adopted, make the notation so clear that reference to it years later will bring the whole subject before the mind in an instant. If this is not done the author of the work himself will not have the patience to dig out the meaning when it is needed.

—E. H. HAGEN,
Appleton, Wis.

Suggests Changing Reference Axis

To the Editor:

JOHN FLODIN'S article "Calculating the Moment of Inertia of Complex Sections," which appeared in the March issue of MACHINE DESIGN, is an article of considerable value to the engineer engaged in analytical work. This particularly is true in the field of aircraft design, where the weight-strength ratio of any

TABLE I

Calculation Table, Assuming Reference Axis near Neutral Axis

Member	Area A	Distance d	Area Moment dA	Second Power Moment d^2A	I_{cg}
Channel.....	3.36	1	3.36	3.36	10.4
Angle.....	2.25	2.92	6.570	19.1844	.67
			9.930		
Plate.....	3	1.75	5.250	9.1875	.0625
	8.61		9.930	31.7319	11.1325
			−5.250	11.1325	
			4.680	42.8644	

Distance between assumed and neutral axis $= \frac{4.680}{8.61} = .5435$

$I = 42.8644 - 8.61 \times .5435^2 = 40.321$

$I/c = \frac{40.321}{3.5 - .5435} = \frac{40.321}{2.95625} = 13.6392$

section is of paramount importance. The variety of sections encountered in such work is so great that the simplified method as outlined in this article relieves a great deal of tedium of calculating moments of inertia, and has proven itself a time saver.

Constant use of various methods of calculating moments of inertia, including the method outlined by Mr. Flodin, led to the adoption of another refinement in the process. Instead of assuming, by inspection, a reference axis near the true neutral axis, it was found desirable to take the reference axis at either the top or bottom edge of the section being investigated. This places the entire area to one side of the reference axis. An example of this is illustrated in the drawing which is shown in the right hand column of this page.

Table II shows clearly that the value of the above assumption lies in the fact that the column headed "Area Moment" now has all items of similar sign, thereby eliminating the subtraction at the foot of the column as well as the two separate additions to determine the plus and minus quantities. This gives not only a simplified proceedure but has its chief advantage in the elimination of possible errors in addition and subtraction; it removes also the danger of error that a reversal of operation necessarily carries with it.

A comparison of the two tables will show how the calculation of an ordinary section can be simplified by assuming the reference axis outside of the section. As the sections become more complex, the elimination of plus and minus quantities in the "Area Moment" column is particularly advantageous in that it removes a likely source

TABLE II

Calculation Table, Assuming Reference Axis at Bottom of Section

Member	Area A	Distance d	Area Moment dA	Second Power Moment d^2A	I_{cg}
Channel.....	3.36	2.5	8.400	21.000	10.4
Angle.....	2.25	.58	1.305	.7569	.67
Plate.....	3	5.25	15.750	82.6875	.0625
	8.61		25.455	11.1325	11.1325
				115.5769	

Distance from assumed to neutral axis $\frac{25.455}{8.61} = 2.9564$

$I = 115.5769 - 8.61 \times 2.9564^2 = 40.323$

$I/c = \frac{40.323}{2.9564} = 13.6392$

of error. In the aircraft industry where the margin of safety over specified strength requirements must, from the standpoint of weight saving, be small, any slight error in the calculated moment of inertia may result in an entirely erroneous conclusion.

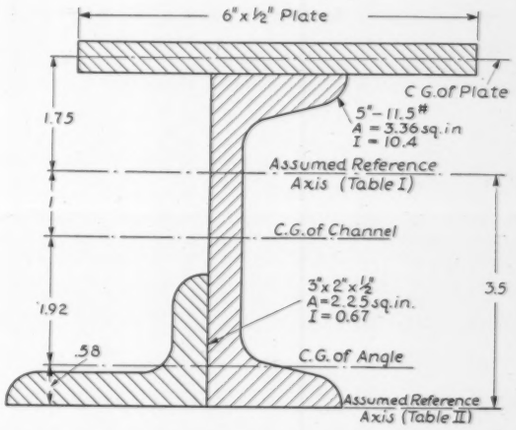
—T. C. BENNETT,
Audubon, N. J.

Is a Bonus System Possible?

To the Editor:

MAY I add a few words to the question of bonus for designers which you raise on page 45 of your April issue?

During the war, bonus was used in some de-



sign work with more or less success. The object, however, was not one of establishing a wage scale, but of speeding up war-time work.

The procedure followed at one plant was this: The chief of the company's estimating department in conference with the chief draftsman and the squad boss or charge man who was to be directly responsible for the design, would agree on what they considered a fair estimate of the cost of the complete plans and the time required for their execution. At the same time the design personnel would be agreed on between the chief draftsman and the squad boss, so that the quality of the design skill available for the job was known fairly well beforehand. Bonus was paid on the basis of a certain sum of money for each day saved, plus a certain percentage (my recollection is 75 per cent) of the money saved in the drawing room, the total bonus being divided prorata among the men. No penalties were exacted because of delay in completing the design. In case a bonus was due because of saving in time, time loss in the shop that was directly traceable to errors in the plans was charged against the design cost at one-half the actual cost of shop time lost.

This plan worked fairly satisfactorily for a

(*Concluded on Page* 55)

MACHINE DESIGN

The True Function of Design

Editorial

NO ONE can observe the trend in the relation of merchandising to production without wondering to what extent some of the experience gained in the field of retail trade can be applied profitably in the manufacture of mechanical equipment. It is believed that if builders of industrial machinery could see their consuming market as clearly as do those manufacturers whose goods pass over retail counters, sales resistance might be lowered.

A FEW years ago most of the articles sold in ordinary department, hardware and novelty stores were created by designers who had little opportunity to know the requirements of the market. The number of articles produced at a given time was largely a matter of guesswork and the price was determined by adding the desired profit to the actual production and other costs. This method of procedure still prevails in many lines of business.

AMONG the first to discard this system were the mail order houses. Gradually they established the practice of investigating the market need for a certain commodity, determining the number that could be sold and the price that should be asked, and then designing and producing the article to these specifications. This plan has proved so successful that it is gaining favor in many industries.

THUS far the idea of making design and production subservient to market considerations has been employed to some extent in the manufacture of automobiles, electric refrigerators, washing machines and other mechanical apparatus capable of being turned out on a mass production basis. Machines made in comparatively small lots do not lend themselves as readily to the new idea as do those consumed in lots of thousands or more. However, we may well ask ourselves if we are not losing some important advantages in our present practice of placing design and production first and marketing and distribution second. Would we not gain immeasurable benefits if our engineers spent more time in the field watching the use of machines and less time over their own blueprints studying what has been done in past design?

WE BELIEVE that an honest survey of the machinery manufacturing industries would show that in a surprisingly large number of establishments the cart is placed before the horse. The tendency to correct this already is apparent. We predict that within a few years the thinking of the industry as a whole will be reversed and that it will be deemed absolutely necessary that design originate from needs of the consumer, and that design, production and sales will be governed by accurate knowledge of the consuming market.

Design and Competition

WHEN business activity lags emphasis on competition is accentuated. Executives are striving to find ways to avoid direct rivalry with competing products. In some quarters, the relation between competition and design is being studied carefully, with the result that gradually the idea that design offers relief from many of the evils of competition is becoming crystallized. If a machine or product can be designed so well that it stands out head and shoulders above its contemporaries, it enjoys a distinction approaching monopoly. If this thought is carried to its logical conclusion, we may say—as Henry Ford has said—that "competition is only bad designing." At any rate, this is a good text for the machinery industries, and particularly for engineers and others responsible for the development of machines, who heretofore have been reluctant to see any connection between design and competition.

Michael Faraday and the Dynamo

Great Moments in Machine Design—Tenth of a series of original drawings prepared exclusively for this magazine symbolizing the designer's contributions to the progress of mankind

NOTEWORTHY PATENTS

A Monthly Digest of Recently Patented Machines, Parts and Materials Pertaining to Design

TO PREVENT end play of horizontal shafts after as well as before wear occurs is the purpose of a new thrust bearing designed by Henry W. Nieman, Bethlehem, Pa. He recently was granted patent No. 1,751,651 by the United States patent office. In the accompanying illustration two embodiments of the invention are shown.

Figs. 1 and 2 show the type applied to the body of a shaft and Figs. 3 and 4 show the thrust bearing designed for use at a shaft end. Describing first the former type, the shaft 10 extends between flanges 12 of the bracket and lies beneath a cross shaft or spindle 13, which has its ends rigidly mounted in these flanges.

When the shaft is rotated the collars 14 move with it, while thrust shoes 16 remain stationary. A lining of any suitable antifriction bearing material may be used as abutting faces of collars 14 and shoes 16 to minimize wear. As wear occurs, however, the light spring 25′, acting through the double cam 25, effects further separation of rollers 23 and wear is taken up. The rollers come to rest against new elements of the involute surfaces 24.

This action is facilitated by the arrangement of four rocker arms 20, arranged in pairs on the respective shafts and each having a rounded lug which fits within the recessed end portion of the corresponding lug 17. These rocker arms are pivotally supported by stud shafts or spindles 19 which have their ends mounted in the flanges 12.

Should there be any expansion of shoes 16 or any tendency for rollers 23 to be brought more closely together simultaneously, this binding or pressure is relieved by part 25 which rotates against the action of spring 26 until the pressure is relieved. Part 25 is so designed that it is readily rotatable by simultaneous action of the rollers in pressing against it. However, it offers a nonyielding abutment when one roller only is active.

The second application of the thrust bearing is shown in Figs. 3 and 4 at the end of a shaft. Action of spring 41, through rollers 42 and 43, is effective in tending to move thrust shoe 34 and cylinder 32 in opposite directions with equal force. Consequently the bearing surfaces of these members are brought into close contact with the thrust surfaces of collar 31 of shaft 30. Any force on shaft 30 which tends to move it to the right tends

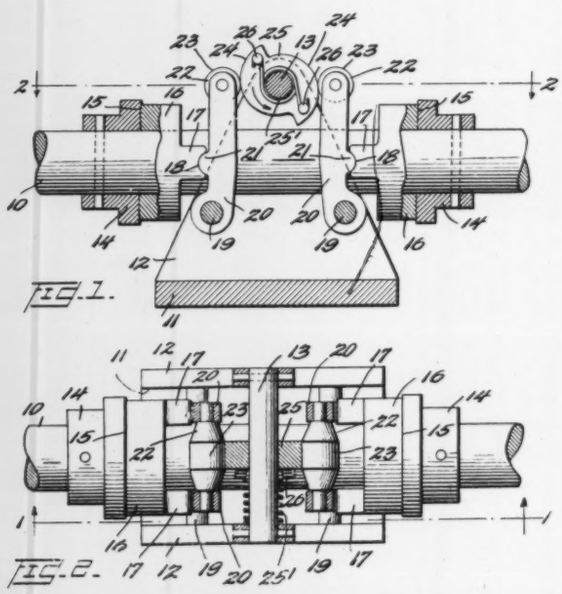
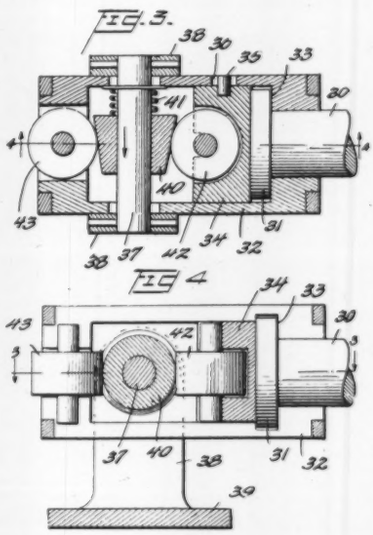


Fig. 1 is a section on the line 1-1 of Fig. 2. Spring 25′ acting through the double cam 25 effects separation of rollers 23. Fig. 2 is a section on line 2-2 of Fig. 1



Two views of the thrust bearing for use at the end of a shaft. Fig. 3 is a section on line 3-3 of Fig. 4 and Fig. 4 is a section on line 4-4 of Fig. 3. Action of spring 41 moves thrust shoe 34 and cylinder 32 in opposite directions

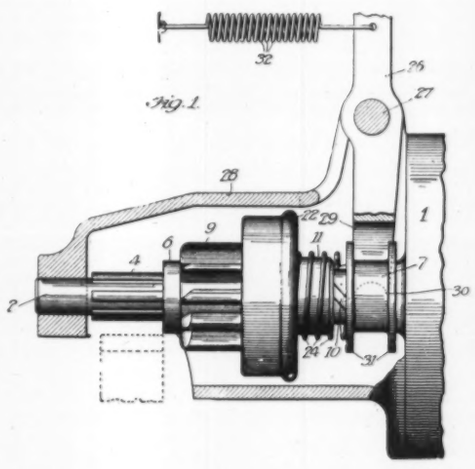
also to draw the cylinder 32 to the right and hence causes roller 43 to bear on the opposite side of the wedge 40.

INVENTION of an engine starter to provide a simple and efficient drive or transmission capable of manual operation for engagement with an engine belongs to William L. McGrath. Patent No. 1,758,233 recently was granted to Mr. McGrath, who has assigned it to the Eclipse Machine Co., Elmira, N. Y.

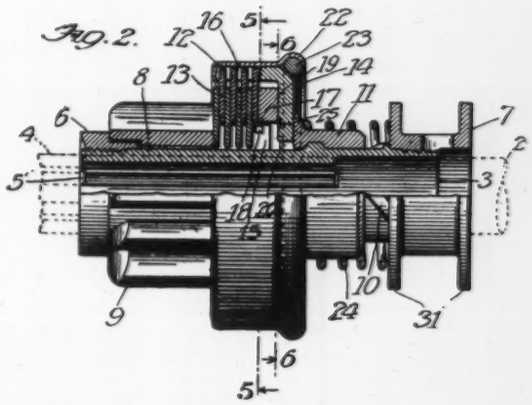
A sectional elevation of an engine starter system including the transmission is shown in Fig. 1, the parts being in normal disengaged position. Fig. 2 is a sectional elevation of the drive proper. Comprising a part of the drive is a shaft which is in the form of a sleeve 3. It is mounted upon the armature shaft for rotary movement therewith and longitudinal movement through a suitable medium, typical of which are splines 4 on the armature shaft with their intervening grooves and the series of corresponding splines 5 with corresponding grooves in sleeve 3.

Control member or nut 11 is connected with the pinion by means of a friction clutch, whereby the torque of the starting motor is transmitted from the screw shaft and the nut to the pinion, thence to the fly wheel of the engine when the pinion is in cranking position. Any suitable shifting means may be employed, which also is capable of operating the usual motor switch. Lever 26 having been actuated, the entire drive assembly slides longitudinally along the armature shaft 2 by reason of the splined connection. In this cranking operation the torque will be transmitted through the screw shaft nut 11, drum 14, washer 17, and friction plates 12 and 13 to the pinion 9.

When the pinion is in cranking position and the starting motor is operated the nut 11 has a screw jack action on the friction plates by reason of screw threads 10. In this operation the spring 24 serves to initiate the above described screw action. When the engine starts on its own power the flywheel rotates the pinion which now becomes the driven member, the screw jack action will become relieved and likewise the frictional resistance between the plates.



Parts are in normal disengaged position in sectional view of engine starter. The drive assembly is restored to normal position by relieving pressure on lever 26



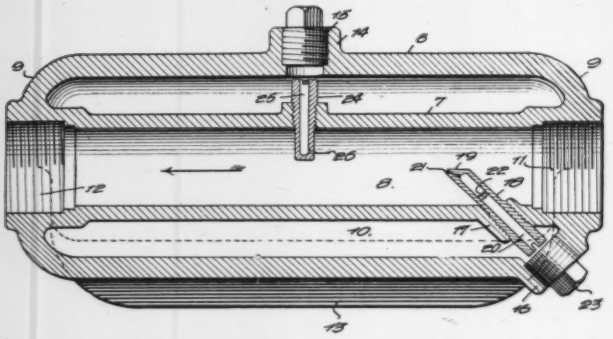
Sectional elevation of drive proper. A friction clutch connects control member or nut 11 *with the pinion. Torque is transmitted from the screw shaft and the nut to the pinion, thence to the flywheel of the engine when the pinion is in cranking position*

By relieving the pressure upon the lever 26 and the shifting means, the drive assembly will be restored to normal position, as by a spring 32, the pinion being thereupon disengaged from the flywheel. In the event of abutting of the pinion teeth and the flywheel teeth, the shifter sleeve 7 and consequently the screw shaft will continue their longitudinal movement to the left, and, owing to the provision of the steep pitch of screw thread, the pinion, whose longitudinal movement has been arrested, will be rotated slightly and sufficiently to result in register between the teeth of the pinion and flywheel. The pinion then will move longitudinally with the screw shaft and go into full mesh with the flywheel.

A LUBRICATOR of the type employed in supply lines or manifolds to deliver oil in sufficient quantity into the air stream has been devised by Omar E. Clark, Denver, Colo. Patent No. 1,755,240 has been granted the inventor who has named the Gardner-Denver Co., Quincy, Ill., assignee. A feature of the lubricator is the fact that delivery of oil is stopped when the supply of air is cut off.

One form of construction is shown in elevation in the accompanying illustration. This particular type of unit has an outer tubular wall 6 and an internal conduit 7, providing an open ended pas-

sageway 8. The walls 6 and 7 are connected at their ends by end wall 9, thereby forming a lubricant reservoir 10 between the walls and around the passageway 8. A filling nipple 14 on the top



End elevation of one form of construction of lubricator. Pressure on the oil in the reservoir causes a small stream to be projected through the port 21 *into the air stream*

of the casing affords access to the reservoir 10. A conduit 20, opening through the lower end of the stem, terminates at its upper end in a port 21 that opens through the nozzle 19 and faces in the direction of the outlet end 12 of the passageway.

In operation, assuming that the reservoir is partially filled with oil, if air under pressure is passed through 8, a portion of this air will find its way through the port 26 and bore 25. Pressure will be created upon the oil, thus causing a small stream to be projected through the port 21 into the air stream. Pressure in the reservoir 10 will be slightly greater than the pressure of the air stream since the port 26 faces the air stream while 21 opens in the direction of it. The difference is sufficient to insure expulsion of oil from the bottom of the reservoir.

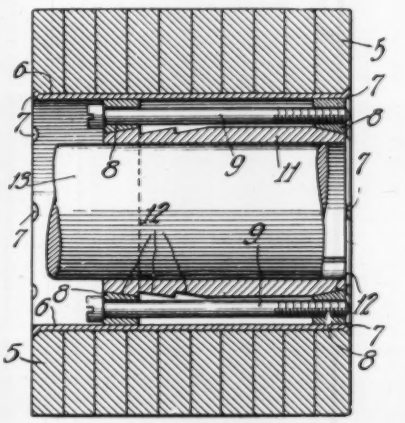
STANDARDIZATION of a pulley having means for readily securing it to shafts of different diameters and lengths is one of the principal features of the invention for which patent No. 1,758,182 was granted to Arthur P. Strong. Dodge Mfg. Corp., Mishawaka, Ind., is the assignee. In the accompanying illustration a diamertic section of the pulley is shown.

The removable hubs are of standard exterior form and dimensions and can be interchangeably assembled with the pulley shell and securing devices, but different hubs have different bores to accommodate shafts of unlike diameters. Laminated wood constitutes the pulley shell 5 which is interiorly fitted with a steel sleeve 6 rigidly secured to the shell by being pinched or upset at intervals as indicated at 7.

Two spaced split clamping rings 8 in the steel sleeve are connected by several machine bolts or screws 9 passing through one of the rings and threaded into the other. Rings 8 have cylindrical outer surfaces and frusto-conical inner surfaces. To facilitate their expansion and contraction each has a number of radial notches of fractional splits 10.

Frusto-conical seats 12 at opposite ends of the hubs are oppositely inclined so that when the bolts or screws are turned to draw the split rings 8-8 toward each other, the rings will be wedged between the outer shell 5 and the hub 11, causing a simultaneous contraction of the hub to gripping the shaft. In the accompanying drawing the left-hand clamping ring is shown applied to the outmost frusto-conical seat 12, but it could be applied to the second or any other surface and the end of the hub may be cut off as indicated by the dotted line to shorten it for use with a shorter shaft without requiring any change in the standard pulley shell or clamping rings.

The construction permits the pulley shell to be mounted upon short as well as long shafts and on



Diametric section of pulley showing the left hand clamping ring applied to the outmost frusto-conical seat 12. *It could, however, be applied to the second or any other surface*

shafts of widely different diameters. Shell and rings are standard and interchangeable and therefore lend themselves to quantity production. Only the hubs need be changed to fit the shell to different sizes of shafts. Also, the pulley shell, the clamping rings, the hub, and the shaft are all mutually adjustable to desired relations, by relatively moving them either axially of the shaft or circumferentially. When adjusted they may be clamped securely and permanently together.

Review of Noteworthy Patents

Other patents pertaining to design are briefly described as follows:

TRANSMISSION—1,757,995. Covered by this patent is a transmission consisting in part of a driving shaft, a lay shaft, a driven shaft, an intermediate shaft, and change spur gears between driving and intermediate shafts. A pair of the gears are mounted on the lay shaft with an internal gear and clutch member and a spur gear and clutch member between the intermediate shaft and

driven shaft. A swinging frame is pivoted on the lay shaft, the driven shaft being journaled into the frame. Assigned to Brown-Lipe Gear Co., Syracuse, N. Y.

GEAR CONSTRUCTION—1,758,692. "A gear having a hub portion provided with a tapered bore and tapered bushing covered with one or more layers of fabric and engaging in said bore." Assigned to Westinghouse Air Brake Co., Wilmerding, Pa.

COVER FEED—1,754,463. Incorporated by this patent is a can feed device comprising in combination a plurality of supporting and separating units, each unit consisting of a plurality of spring pressed blocks retractible to permit the passage of can ends, and means for releasing the units and for securing them in desired positions relative to the can ends. Assigned to American Can Co., New York.

CLUTCH—1,754,794. A patent has been granted for a clutch which has in combination "a constantly revolving driving pin, a retractible clutch block adapted to be engaged by the driving pin, the clutch block having an inclined face, a bracket, an arm pivotally mounted on the bracket, and a shoe having an inclined face carried by the arm." Assigned to Verdin, Kappes & Verdin Co., Cincinnati.

ADJUSTING MECHANISM—1,758,231. This patent was granted for a bed-adjusting mechanism for stamping presses and covers a sheet metal working press having a bed and means for adjustably mounting a lower die or tool relative to the bed comprising a lower slide adapted to carry the die and guided by the bed, and means for adjusting the die vertically relative to the bed. Assigned to Niagara Machine & Tool Works, Buffalo, N. Y.

GEAR BOX BRAKE—1,753,384. This invention embraces "a power transmission for lathes, including means for driving a lathe spindle-rotating mechanism, toothed gear means for driving a mechanism for axially shifting the lathe spindle, and means co-operating with last mentioned driving means to retain the same against movement." Assigned to Merritt Engineering & Sales Co. Inc., Lockport, N. Y.

DRAFT GEAR—1,775,429. Involved in this patent for a draft gear is the combination of a pair of opposed followers, two sets of co-operating and wedging and friction elements interposed between the followers, the friction elements being bifurcated to provide a spring chamber located on the horizontal medium portion of the gear. Means are provided for resisting the wedging action of the elements. Assigned to Union Draft Gear Co., Chicago.

CENTRIFUGAL PUMP—1,757,281. "In a centrifugal pump the combination with a casing, a pair of removable rings mounted in said casing and formed to provide an annular impeller channel, an impeller wheel operating in said casing, impeller blades at the periphery of the wheel operating in the channel, said channel having peripheral intake ports arranged to deliver fluid into said channel and into contact with the blades, and means blocking direct radial flow between the inlet and the impeller wheel pockets." Assigned to Auto Prime Pump Co., Cleveland.

PROFESSIONAL VIEWPOINTS

(*Concluded from Page* 49)

time, at least in so far as the designers were concerned: it acted as a stimulus both to speed up the design work and to "be sure you are right before you go ahead." But the increasing turnover because of the turmoil of the war soon made it impossible to foretell what quality of design talent would be available for any given job, and the plan had to be abandoned. Economically its great weakness is that there is little advantage in having the drawings ready ahead of schedule time, except possibly in cases of jobs of such importance that other work is made to yield the right-of-way in the shops, as was often the case during the war.

In another plant, where the scales to be used in different design situations were established as a part of the company's design standard, a bonus was paid on the basis of square feet of drawing area completed per week. This is an excellent plan where drawing is all *drawing*, or tracing, but it is hardly applicable to design, as pointed out by John F. Hardecker in your April issue. This plant lost many of its best men because they were unwilling to work under such conditions—fortunately.

Any bonus applied to design work, no matter how successful it may appear to be at first, almost certainly would be injurious in the long run. The man who is a designer at heart, who has the intelligence, the experience, and the interest that make a designer of the highest caliber, would not as a rule reap a just reward under a bonus system. He is likely to be the sort of man who willingly spends an hour or a day to help a fellow worker out of a difficulty, or to straighten out a tangle, not because he is necessarily a good-hearted mollycoddle who cannot say "no," but because he sees the design problem in a larger sense than does the man whose major interest is the bonus. From the professional point of view I feel that the notion of bonus for speed borders on the insulting. What would you think of the ethics of a physician who said to you, "It would normally take you a month to recover from your illness, but I can speed that up to three weeks if you pay me a 25 per cent bonus?"

—JOHN FLODIN
Minneapolis.

Additional facilities for industrial research as well as more commodious quarters for all departments will be available in the near future at the Mellon Institute of Industrial Research.

MEN OF MACHINES

Personal Glimpses of Engineers, Designers, and Others Whose Activities Influence Design

IN ADDITION to engineering activities, W. S. Rugg, vice president in charge of engineering, Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa., has been placed in charge of sales activities. A native of Wisconsin and a graduate of Cornell university, he has been associated with the Westinghouse company since 1892. After doing engineering and sales work in the Chicago office, he was transferred to New York where in 1909 he was made manager of the office. In 1917 Mr. Rugg was recalled to East Pittsburgh to become manager of the railway department, and shortly after, the marine department. He became assistant to H. D. Shute, then vice president of sales, in 1920 and in 1922 was made general sales manager, after which he was appointed vice president in charge of engineering in 1925.

HOLDING a long record of mechanical achievement, Ambrose Swasey stands out today as one of the most eminent engineers in the history of design and invention. Besides being an engineer and manufacturer he also is a traveler, philanthropist and author. Born at Exeter, New Hampshire, Dec. 19, 1846, his schooling was completed in a district schoolhouse, but nevertheless he holds several honorary degrees. In 1880 with W. R. Warner, Dr. Swasey came to Cleveland to establish the foundation of a company which has gained prominence throughout the world for its machine tools and precision instruments. At the recent celebration of the fiftieth anniversary of the Warner & Swasey Co., Dr. Swasey paid fitting tribute to Mr. Warner who died a year ago.

ELECTION of B. F. Waterman as president of the American Gear Manufacturers association places at the head of that organization a man of 34 years experience in gear design, production and application. During the entire period he has been with Brown & Sharpe Mfg. Co., Providence, R. I., having entered as an apprentice upon completing his education in the Providence public schools. Leaving the machine shop he subsequently was assistant foreman and foreman of the gear department. Later he entered the engineering department where he now holds the title of gear engineer. Mr. Waterman has been chairman of the general standardization committee almost continuously since the inception of the gear manufacturers' association.

ASSISTANT vice president is the new title which has been bestowed upon S. M. Kintner, director of Westinghouse research laboratory. Simultaneously he assumed engineering department duties under the direction of W. S. Rugg. Mr. Kintner, who is well known for his research and engineering work, was graduated from Purdue university in 1904. For a time he was a member of the faculty of Western University of Pennsylvania, now the University of Pittsburgh. After he became associated with the Westinghouse company he was connected with the railway engineering department in charge of alternating current and railway motor design. He joined the Electrical Signalling Co. in 1911, and in 1920 he again became associated with the Westinghouse organization.

CO-OPERATING with Engineering Foundation in alloys of iron research, the Battelle Memorial institute, Columbus, O., is particularly fortunate in having Dr. H. W. Gillett as its director. His wide knowledge of metallurgy has been gleaned over a period of years since he concluded his graduate work in chemistry at Cornell university in 1910. During his 12 years with the bureau of mines he studied many phases of the science. In 1924, he became chief of the metallurgical division of the bureau of standards and took an active part in the supervision of research in many lines of metallurgy. Dr. Gillett became director of the Battelle institute in 1929 and besides maintaining contact with various technical societies and working on many of their metallurgical committees, he includes in his work for the institute the task of editorial director of the new journal, *Metals and Alloys*.

TEN members, one of whom is Charles F. Kettering, form a committee appointed to select the United States citizen who in 1929 performed the most distinguished service to social

Leaders in Design, Engineering and Research
AMBROSE SWASEY
W. S. RUGG
B. F. WATERMAN
S. M. KINTNER
DR. H. W. GILLETT
C. F. KETTERING

and economic welfare of the engineering profession and humanity at large. Mr. Kettering is prominent as an inventor and is vice president and director of the General Motors Corp. The award, in the form of a gold medal named after Henry W. Clausen, will be presented each year by the American Association of Engineers. After his graduation from Ohio State university in 1904, Mr. Kettering taught school, later entering the employ of the Star Telephone Co., Ashland, O. Subsequently he was with the National Cash Register Co., Dayton, O. and Dayton Engineering laboratories. Many patents and engineering developments have come from his farsighted scientific vision.

* * *

Charles F. Wasserfallen has resigned as vice president in charge of manufacturing of the Oakes Division of the Houdaille-Hersey Corp., Detroit. He formerly was chief engineer and works manager of the Detroit Carrier & Mfg. Co. which now is a division of the Houdaille-Hersey Corp.

* * *

Clarence L. Collens, president of the Reliance Electric & Engineering Co., Cleveland, was elected president of the National Electrical Manufacturers association at the recent spring meeting at Hot Springs, Va.

* * *

G. M. Eaton recently was appointed director of research of Spang, Chalfant Co., Ambridge, Pa. He formerly was identified with Molybdenum Corp. of America, Pittsburgh.

* * *

Dr. Albert F. Zahm, chief of the division of aeronautics in the Library of Congress, has been awarded the medal presented annually by Villanova college for distinguished service in advancement of science.

* * *

Kenneth M. Ronan has been elected president of the Warrior Aeronautical Corp., Warren, O., successor to Alliance Aircraft Corp. Mr. Ronan formerly was chief engineer of the Stinson Aircraft Corp., Detroit.

* * *

John R. Replogle has been named chief engineer of Copeland Products Inc., Mount Clemens, Mich. He formerly was chief engineer and one of the organizers of Nizer Corp., which merged about 1926 with Kelvinator, and developer of the Nizer compressor. In 1919-1920 Mr. Replogle was chief engineer of Frigidaire and has been with Remy Electric and General Motors laboratories and Western Electric Co. in engineering and research capacities.

* * *

R. J. Allen has been appointed research engineer with the Worthington Pump & Machinery Corp., Harrison, N. J. Mr. Allen was associated with Rolls-Royce Co. of America, Springfield, Mass., for 12 years.

* * *

LeRoy S. Schell has been appointed designing engineer in charge of transformer engineering at the Erie plant of the General Electric Co., being transferred from the engineering department of the Pittsfield plant, where he has been for the past 12 years.

* * *

A. Francis Arcier has been appointed chief engineer of the Waco Aircraft Co., Troy, O., succeeding Russel F. Hardy. Mr. Arcier was engineer for Handley Page Ltd., during the World war and was associated with the designing of the Handley Page bomber and many other well known machines.

* * *

Erwin G. Bailey, president, Bailey Meter Co., Cleveland, was awarded the Edward Longstreth medal by the Franklin institute, Philadelphia, May 21. He received the honor upon the recommendation of the Institute's committee on science and arts in recognition of his numerous inventions and developments of regulating and controlling devices, measuring and recording instruments, etc.

Obituaries

Edwin C. Bruning, vice president of Charles Bruning Co. Inc., manufacturer of engineers' supplies, instruments and equipment, New York, died May 25. He was the eldest son of Charles Bruning, president of the company.

* * *

Capt. L. M. Woolson, prominent automotive designer, met death when his plane crashed during a blinding snowstorm in New York state late in April. Outstanding among his achievements was the design and development of the Packard aircraft diesel described in the April issue of MACHINE DESIGN. A picture and short biographical sketch of Capt. Woolson also appeared in that number.

PUT PUT AND TUT TUT

What the fish thinks about the invasion of his watery home by the put-putting outboard motor nobody knows. Chances are it wouldn't be fit to print.

What the owner of the outboard motor thinks when it refuses to put-put he makes no bones of letting the world know. And it's equally unfit to print.

But, oh, the "grand and glorious feeling" when the old boat is hitting evenly as she swiftly skims over the waves, kicking up the spray with her heels.

And the development of the outboard motor from the toy it used to be to the splendid power plant it now is, has been due in large measure to the use of Cold Finished Steel Products.

For the superior strength and dependability to stand the strain of the record-smashing speed that this sport demands —*no other material takes the place of Columbia Cold Finished Steel Bars.* Columbia Steel and Shafting Co., Pittsburgh, Pa.

COLUMBIA

COLD FINISHED BARS AND SHAFTING

TOPICS OF THE MONTH

A Digest of Recent Happenings of Direct Interest to the Design Profession

TWENTY-FIVE years of progress in automotive design and development was celebrated at the quarter-century anniversary meeting of the Society of Automotive Engineers held May 25-29 at French Lick, Ind. An historical exhibit climaxed by a pageant was a center of interest and brought out the tremendous advancement in automotive engineering since the inception of the society.

Twelve technical and two general sessions were characterized by unusually interesting and informative papers on various subjects covering a range of phases of automobile manufacture. While exhibits indicated the tendency toward new trends in design, many features of modern engine practice could be found among old models shown. The Manly engine was an example of this.

Vincent Bendix, president of the Bendix Aviation Corp., Chicago, is the nominee for the presidency of the society for 1931.

* * *

Mechanical Movements on Exhibition

An interesting exhibit embodying more than 200 mechanical movements, machine models and combinations of driving mechanisms with models of automotive and other machinery was opened to the public May 13 by the Newark, N. J., museum of art, science and industry. The displays, recently acquired by the museum, portray every form of motion in the models shown including transmission of power, measurement of power, generation of power, hydraulics, manufacturing, navigation, motion controlling devices, horological timing devices, a complete story of design.

* * *

Discusses Developments in Gun Construction

Advantages of making steel castings by the centrifugal method were explained by Brigadier General T. C. Dickson, commanding officer at Watertown arsenal, to members of the Cleveland Engineering society at a recent meeting. In addition to this process, as applied in the construction of large guns, the employment of cold working has been found particularly valuable.

The centrifugal casting machine works the air to the center and segregates the carbon at the bore, thereby producing a steel gun barrel of quality grain structure. Hydraulic pressure to increase the size of the bore raises the elastic limit and comparisons drawn by General Dickson showed increased efficiency in this process over the built-up method. The value of X-rays in examining welds has proved a boon to fabrication of ordnance equipment at the arsenal and has done much to improve welding both in point of workmanship and best suited types for particular applications.

* * *

Mining Machines Reveal Design Trends

Far-reaching advances in the development of machinery for use in the coal mining industry were portrayed at National Exposition of Coal Mining Equipment held recently in Cincinnati. Over 100 exhibitors displayed all types of mining machinery. The demand for floor space exceeded previous records, this being due largely to the interest in mechanization which has been reached by the industry.

An outstanding feature in connection with the exhibits was the large number of coal loaders on display, ranging from single loading units to continuous operation machines incorporating coal cutting, gathering, conveying and loading mechanisms. The latter, particularly, typified the increasing tendency toward "straight line" operation in not only this but practically all industries.

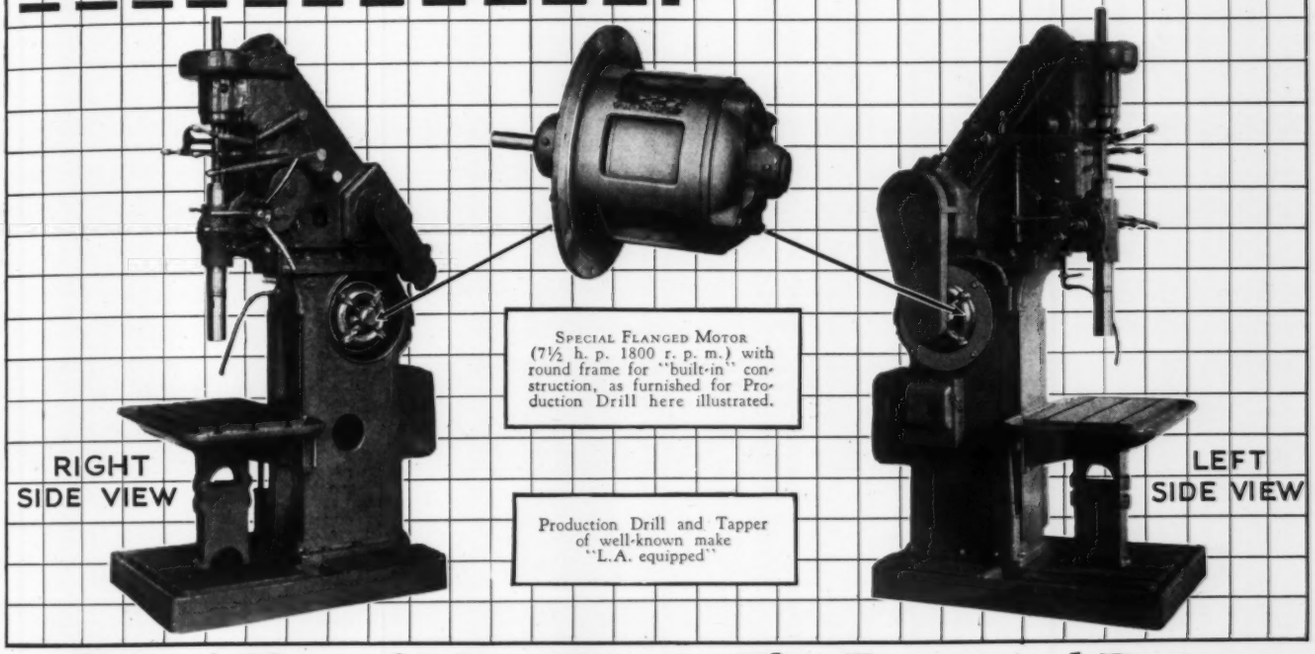
* * *

Recommend National Gage Standards

Establishment of national standards for pressure and vacuum gages was recommended at a general conference held in New York May 15 under the auspices of the American Standards association. Thirty-five representatives of manufacturers and users of pressure and vacuum gages and of technical, governmental and safety bodies having an interest in such gages were present.

In accordance with the recommendations of the conference, the scope of the technical committee's work will include nomenclature and definitions,

STEPPING UP MACHINE EFFICIENCY WITH "Custom Built" Motors


RIGHT SIDE VIEW
Special Flanged Motor (7½ h. p. 1800 r. p. m.) with round frame for "built-in" construction, as furnished for Production Drill here illustrated.
Production Drill and Tapper of well-known make "L.A. equipped"
LEFT SIDE VIEW

Special Flanged Motor Permits This Economical Design

THE trend today in machine design is toward simpler and more economical methods of applying power. Machinery builders are designing new machines and redesigning old ones to eliminate unnecessary parts, friction, and power waste—STEPPING UP MACHINE EFFICIENCY—to meet the need for rapid, low cost production. That is why more and more machinery builders are using Louis Allis "custom-built" motors, and selling machines with motors built in as integral parts.

The new Production Drill shown above is an excellent example It is equipped with a Louis Allis special flanged motor "built for the job." The left side view shows how the flange is bolted to the column making the motor an integral part of the drill The right view shows how the machine is designed to receive and to contain the motor Here are the advantages of this modern method of power application:

1. Standardized Motor Mounting Eliminates Engineering and Delay in adapting machine to motor.
2. Simplicity—Saves Cost of line shafts, belts, pulleys, idlers, and motor mounting brackets.
3. Installation Cost Minimized — complete machine with motor installed as a unit.
4. Compactness—Saves Floor Space.
5. Motor protected from dust, dirt, grease, and flying chips.
6. Dependable Speeds—No Belt Slippage.
7. Motor Easily Removed from column by electrician for inspection or servicing.
8. Motor Accessible for Greasing.
9. Maintenance Cost Reduced—no belts to repair or replace.
10. Saves Power otherwise wasted driving line shafts, belts, and other unnecessary parts.
11. SALEABILITY INCREASED—accepted as a modern production machine.

Are you considering designing a new machine, or redesigning an old one? Are you striving to increase the efficiency of machines that you build? In any event it will pay you to investigate L.A. "custom-built" motors. Louis Allis engineers will gladly assist you in selecting or designing motors exactly suited to your needs. This L.A. service is yours for the asking.

Write for Bulletins describing L.A. "Custom-Built" Motors.

Prompt Deliveries *Nationwide Service*

LOUIS ALLIS

"CUSTOM-BUILT" *Electric* MOTORS

THE LOUIS ALLIS COMPANY
MILWAUKEE, WIS.

Motor Specialists for 28 Years *Offices in Principal Cities*

rules and specifications for installation and use, method of testing, method of expressing allowable errors, accuracy requirements, capacity ratings, connections, indicator hands and stop pins, dials and graduations, bezel rings and attachments, case sizes and mounting holes.

The conference favored, in general, the development of specifications which would tend to unify the external features of gages of the indicating types, and permit a reasonable amount of interchangeability between the various makes.

* * *

Design Has Important Role in Machine Tools

Improving design to overcome sales resistance was one of the important topics discussed at the twenty-eighth spring convention of the National Machine Tool Builders' association at Columbus, O., May 19 and 20. Design as a means of getting out of the rut of competition and a thorough study of the market for machine tools were emphasized. Developing the product to fit the requirements in the market will be one of the big factors in solving a vital problem of the machine tool industry, it was brought out. The policy of the association for the next three years will be to devote its primary efforts to sales and sales promotion. In carrying out this aim design is going to play an important role.

* * *

Patent Bill Drawn to Prevent Fraud

Congress now is considering the Crampton bill which has been drawn up to prevent fraud, deception and improper practice in connection with business before the United States patent office. As presented this bill provides that it shall be unlawful for any person to practice as a patent agent or patent attorney unless he is fully recognized to practice before the patent office; to conduct or maintain in connection with any other person, an office of any kind for preparing or prosecuting applications for patents; or to act as a counselor with respect to matters relating to patents. Exception was made as to clerks or others supervised by persons duly qualified.

* * *

Steel Show To Remain at Stevens Hotel

Definite announcement recently was made that the twelfth National Metal exposition will be held at the Stevens hotel, Chicago, Sept. 22-26. Although the Stevens hotel originally was designated as the place for the show, later consideration indicated that Soldier Field might be used because of the larger amount of space available. The arrangement of space at Solider Field did not prove satisfactory, however, according to W. H. Eisenman, secretary of the American Society for Steel Treating, the organization sponsoring the exposition and the National Metal congress to be held in conjunction with it.

* * *

James Watt Is Named Greatest Engineer

In a poll of the deans of engineering colleges recently taken by the school of technology of Villanova college, of which Carl T. Humphrey is dean, James Watt, the inventor of the steam engine, was placed at the top of the list of the five greatest engineers of all time. Second came Leonado di Vinci, builder of the Martesana canal, third Thomas A. Edison, then James B. Eads and fifth Ferdinand de Lesseps. Of the ten greatest engineers of the past 25 years, the poll placed President Herbert Hoover at the head of the list for his work in mining and engineering.

* * *

Human Factor in Design Commands Interest

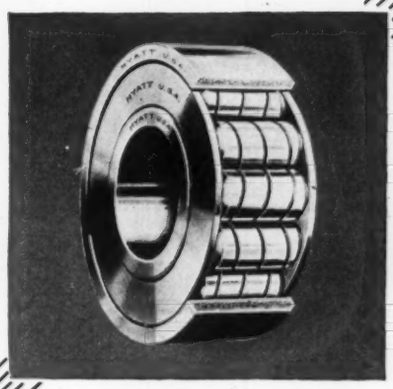
That consideration of the human element in design of machines extends beyond the interest of the engineering profession is evidenced by the article in the Science and Invention department of the May 31 issue of *Literary Digest*. It is an excellent abstract of "The Time Is Ripe for Engineers to Consider the Human Factor," the first of the series on that subject published in the February number of MACHINE DESIGN. Recognition of this important cause is gaining and will do much to impress the fact that cognizance in this direction will be the answer to a great deal of the criticism of the machine age.

* * *

Textile Exhibition Indicates Development

Progress in design of textile machinery was one of the outstanding characteristics of exhibits at the Knitting Arts exhibition held May 12-16 at the Commercial museum, Philadelphia. Variable speed motor drives, automatic speed reduction for lace clocking operations on leggers, single needle sections for full-fashion leggers above 42 gage, new rocker motions, automatic loose course and extra heavy draw cams and new type shogging cam were some of the features of the new trend in this type of equipment.

A new looping table equipped with gear reduction unit in place of the previous chain reduction, and an oilless bearing in the transmitter also was shown. Greater adaptability of aluminum cones for various types ot winding machines and a larger humidor for seamless hosiery cones was another innovation brought out in the exhibit of another company. The tendency toward increased production through more efficient and better designed machines was indicated.



ALL your mechanical equipment needs the kind of protection that Hyatt Roller Bearings afford.

Longer bearing life, smoother operation, capacity to withstand punishment and unusual strains, freedom from constant oiling, adjustments, and shut downs for bearing repairs — these are the advantages of Hyatt protection.

The Hyatt Roller Bearings that are in service on practically every type of equipment today are proving the worth-while economies effected by these better bearings.

Specify Hyatt Roller Bearings on your next order for new equipment. Hyatt protection, you will find, is a profitable investment.

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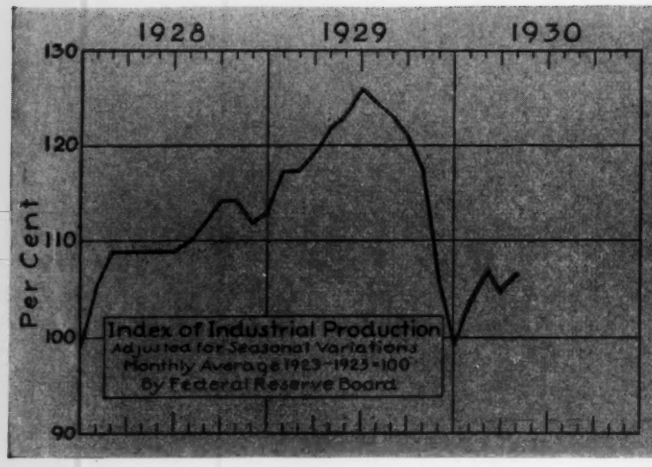
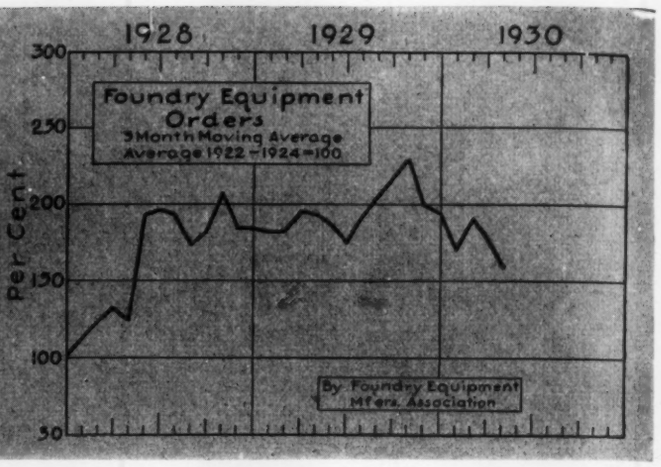
PRODUCT OF GENERAL MOTORS



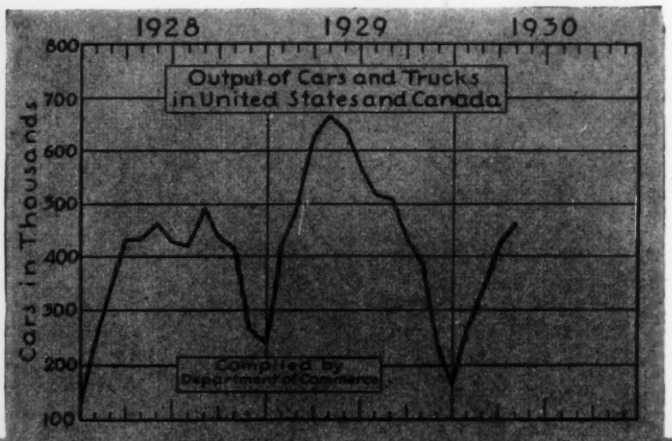


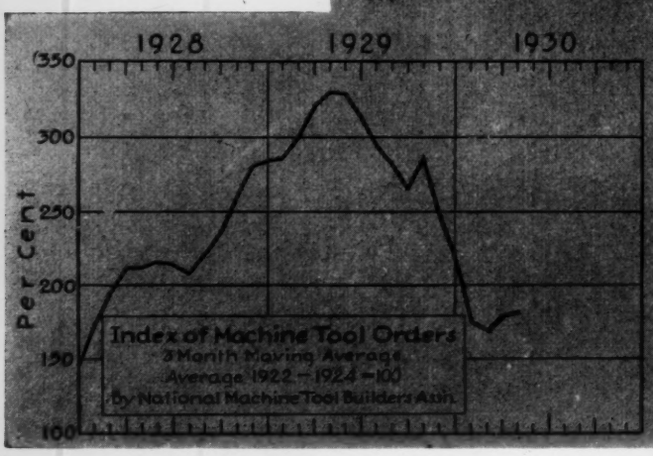
How Is Business?

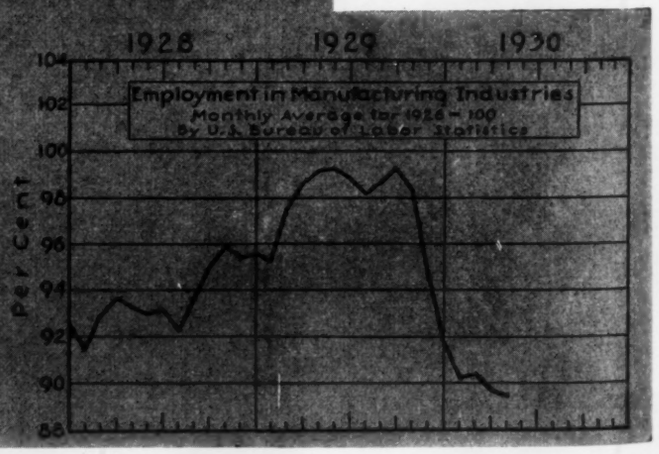
INDUSTRIAL market conditions point toward a summer of continued dullness, despite strengthening of basic economic conditions. Tangible improvement, however, should begin to appear in the autumn, perhaps gaining momentum as the year draws to a close; but normal volume of business may not develop before the middle of next year if previous recovery periods are taken as a guide.

With machinery sales for May, 25 to 40 per cent of last year's figure for the same month, the market still is limited. In the New York and Chicago districts several large orders have served to hold up the rate. Export orders continue fairly steady and textile machinery builders report a spotty demand, with most sizable orders coming from southern mills. Foundry activities indicate some increase in production but machine demand from this source is expected to be curtailed. Engine builders bought heavily last year and generally have equipment to carry them for some time. The same is true of refrigeration equipment builders. The employment index shows further decline.

In the iron and steel industry the seasonal letdown becomes more accentuated with the price situation still unsettled. Quotations on soft steel bars, one of the most widely used finished steel products, recently dropped $4 a ton or more than 10 per cent since last October. Automotive requirements are slackening. Copper is 13.00c and yellow brass rods (high, full turnings) 18.00c. Spring wire is 3.30c and all sizes of machine bolts, cut thread, now are quoted 73 per cent off list.

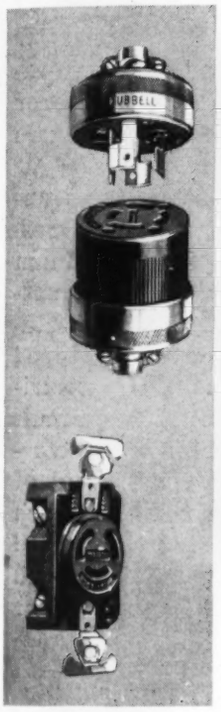


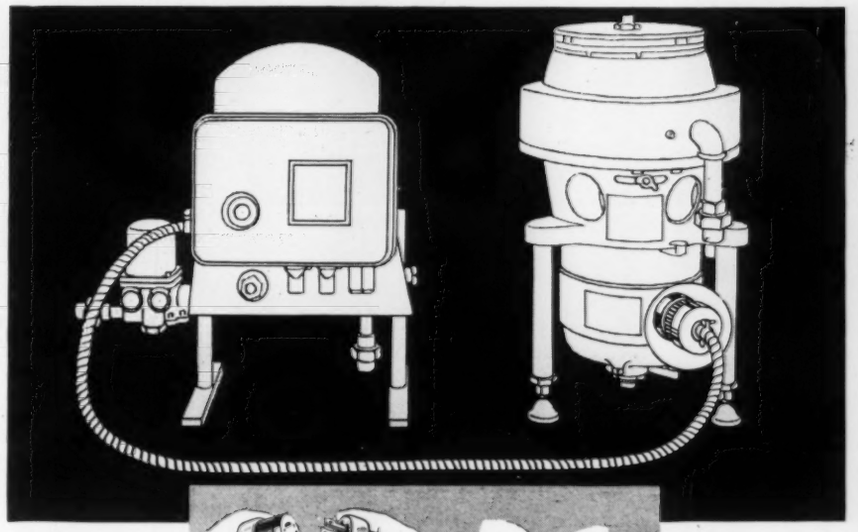


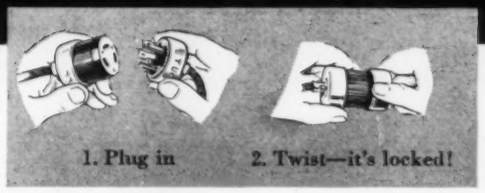


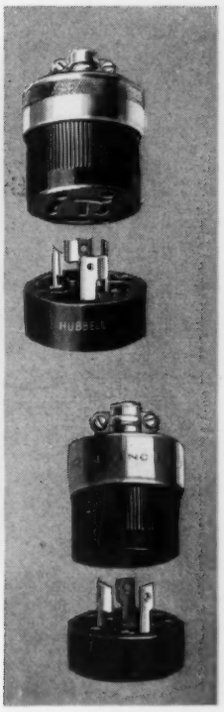
So the Chief Engineer

. . . satisfied the demands of both Convenience and Fire Underwriters by specifying a Twist-Lock Device for this Oil Burner.









This oil burner is installed in the furnace ash pit with a control box outside. Convenience demanded that the electrical connection between the two units be separable; however, the Fire Underwriters required it to be secure. By simply specifying a Twist-Lock Motor Plug, the Chief Engineer satisfied both connection requirements with ease and economy.

Twist-Lock Devices are separable; a twist of the wrist unlocks them. Yet they possess the security of a direct connection, since they will not part . . . even under severe vibration and hard pull on the cord.

To gain the advantage of disassembly convenience, with absolute security which defeats disconnection trouble, makers of all kinds of electrically operated and motor-driven machines are incorporating Twist-Lock Devices. The three and four wire types also provide for proper grounding; thus protecting the operator of the machine against electrical hazard.

These are advantages which appeal to the buyer of a machine. Make them sales features of the equipment you build. The extra cost is negligible. Let us tell you about it. Use the coupon, now.

HUBBELL *Twist-Lock* Devices

Mail Coupon to nearest office. *Main Office: Bridgeport, Conn.*

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138 Marietta St.
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Please send information on Hubbell Twist-Lock Devices

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M. D. 6-30

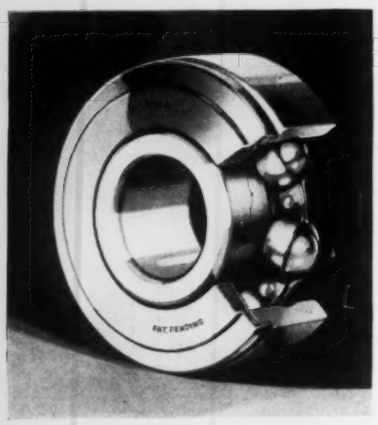
City and State_____

NEW MATERIALS AND PARTS

Worthy of Note by Those Engaged in the Design of Mechanisms or Machines

Develops New Series of Bearings

ANNOUNCEMENT has been made by the Norma-Hoffman Bearings Corp., Stamford, Conn., of the development of an entire new series of "Greaseal" bearings which have been made up for the purpose of reducing costs of machining and mounting by eliminating a number of housing parts. The series is a combination of the usual standard annular closed type of bearing employing a seal feature that consists



Cutaway view of bearing showing construction which consists of a seal feature including steel side plates with felt interposed. The extension of the inner ring completes the seal

of steel side plates with felt interposed, closing one side of the bearing.

Extension of the inner ring completes the seal, which makes this combination particularly effective in retaining the lubricant contained in the bearing, as well as in preventing contamination of the lubricant by dirt from the outside. All-steel construction, load carrying capacities equal to those of the regular series, the standard width of the outer ring, and the added seal which provides for large internal grease capacity, make this series of bearings especially desirable for both high and low speeds.

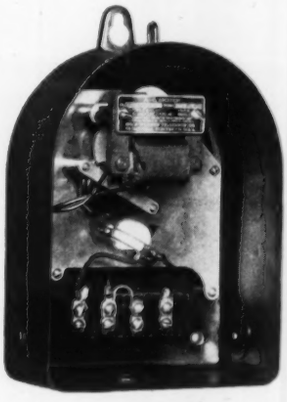
Units Are Indoor Cubicle Type

INDOOR cubicle type switching equipment comprises a new line recently announced by Westinghouse Electric and Mfg. Co., East Pittsburgh, Pa. The cubicles, consisting of oil breakers and vertical swinging panels, with disconnecting switch compartments, may be used for the control of feeders, transformers, or generators, or for any other application calling for oil breakers and a switchboard.

These cubicles are designed to give the advantage of complete factory assembled, steel enclosed, safety type switching equipment at a minimum expense. Units are for either manually, alternating current motor, or direct current solenoid-operated oil circuit breakers and may be equipped with automatic reclosing equipment, if electrically operated, which will reclose the circuit breaker a definite number of times on a predetermined schedule. The cubicles are constructed of structural steel members and one-eighth inch thick, open hearth, steel shapes securely welded together to form a rigid structure unit that will not warp due to changing temperatures or handling.

New Time Switch Is Automatic

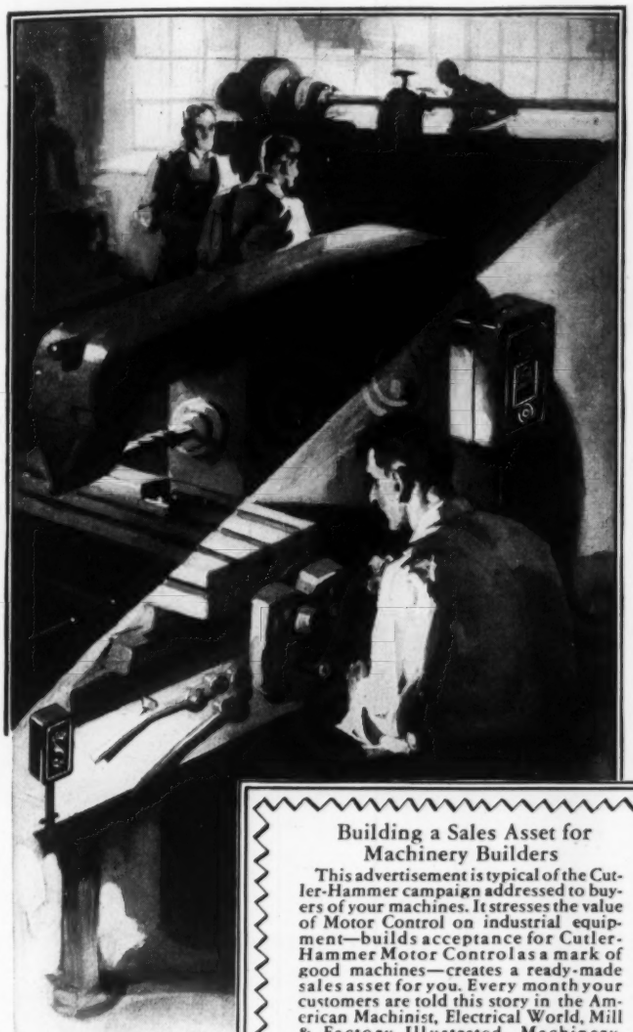
FOR closing and opening electric circuits at short predetermined intervals and on a uniform schedule, a new automatic time switch recently introduced by the General Electric Co., Schenectady, N. Y., has a wide and varied ap-



Open view of new automatic time switch in which the contact-making device is driven by a synchronous motor. A sheet metal dust-proof case is provided for mounting and the front cover is hinged

plication. Designated as type TSA, it automatically energizes a circuit to control intermittent operation of electric equipment. The time cycle may be a few minutes or several hours, and is

YOU CAN JUDGE A MACHINE *by* WHAT ITS MAKER BUYS FOR IT



Building a Sales Asset for Machinery Builders

This advertisement is typical of the Cutler-Hammer campaign addressed to buyers of your machines. It stresses the value of Motor Control on industrial equipment—builds acceptance for Cutler-Hammer Motor Control as a mark of good machines—creates a ready-made sales asset for you. Every month your customers are told this story in the American Machinist, Electrical World, Mill & Factory Illustrated, Machinery, Factory & Industrial Management. An interlocking campaign also appears in the Saturday Evening Post, Nation's Business, Time and Electrical World.

●

PARTS which the builder makes himself . . . parts which he buys from others . . . the parts making up any machine can be classified this way. Of the two groups, the purchased parts are more important . . . not necessarily in the operation of the machine . . . but in judging it before you buy it for your production work.

The builder's ideal of quality for the whole machine is reflected by the parts he uses in it. If, for example, he incorporates a part which is known to have exceeded every test Industry can give it, such a unit marks the machine as having been designed and built throughout to meet your requirements, regardless of cost.

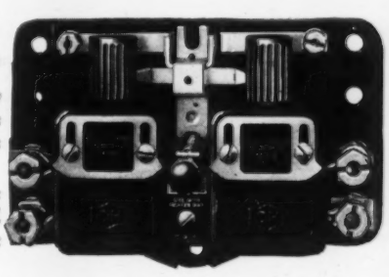
Cutler-Hammer Motor Control is a built-in part on more and more motor-driven machines. It is evidence that the builders of these machines have carefully weighed three factors of utmost importance in machine service and performance . . . the *accurate* protection of motors so bigger loads can be handled with safety . . . the protection of men and the machine itself . . . the savings of steps and time. Thus Cutler-Hammer Motor Control is even more than evidence of quality throughout a machine. It is evidence the machine will measure up to your requirements.

Cutler-Hammer Motor Control is also available for all motors you purchase separately. It is recommended by leading motor builders and stocked by reliable electrical wholesalers in principal cities.

MORE USABLE H.P.

The Cutler-Hammer Thermal Overload Relay protects motors so accurately against overloads that heavier loads are handled with safety. And its accuracy is permanent.

Nothing to replace after tripping . . . just press reset button to put motor back in service. It is used on the entire C-H Standard Line of Automatic Starters.



CUTLER-HAMMER, Inc.

Pioneer Manufacturers of Electric Control Apparatus

1310 St. Paul Avenue
MILWAUKEE, WISCONSIN

CUTLER HAMMER

The Control Equipment Good Electric Motors Deserve

(A-230)

repeated continuously. Owing to the short time of a cycle—one complete "on" and "off" period—the operation is without regard to the time of day.

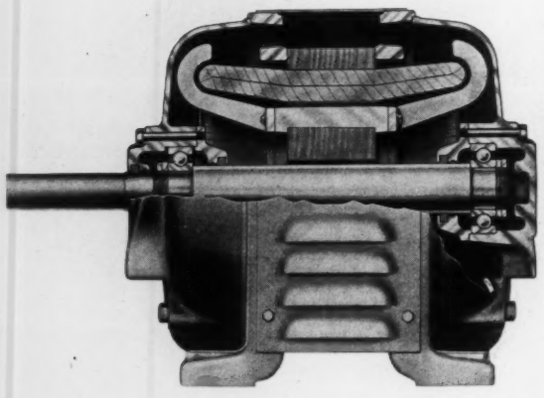
Time cycle depends upon the gearing used, and is fixed in each case at the factory. In certain forms, however, the relative amount of time "on" during the complete time cycle can be varied by the user. Switches are single-pole, with one-way or two-way contacts, and are rated 5 amperes at 115 volts or 2½ amperes at 230 volts. On circuits of higher ampere rating a magnetic switch can be employed in conjunction with the time switch.

The contact-making device is driven by a Warren "Telechron" synchronous motor. The device is housed in a sheet-metal, dust proof case for indoor mounting, with knockouts and a hinged front cover that is secured by a latch at the top, which may be sealed. A hanging lug is provided at the back for wall mounting.

Enclosed Motor Has New Features

INTRODUCTION of an enclosed motor that embodies several new features has been announced by U. S. Electrical Mfg. Co., Los Angeles, Calif., and is known as the "Uniclosed" motor. It has a pressed steel housing, dual fan blast, embodies the "AutoStart" principle, asbestos protection and is equipped with a solid die cast one-piece aluminum rotor. In this advanced design the active unit is shielded. Protection is provided for moving parts and for the electrical windings.

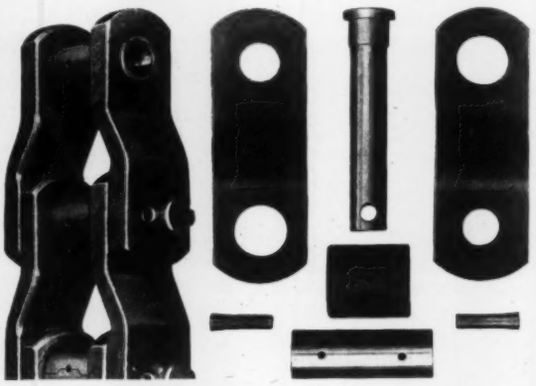
Ventilation is furnished by the employment of a dual fan blast and dual exhausts. Two air streams are drawn through the motor from both ends by high capacity fans located on opposite sides of the rotor. Each fan creates separate air currents which meet at the middle of the frame where the heated air is expelled on both sides of the motor through dual exhaust louvers. There is no direct exposure of the windings and internal parts are shielded from water and against abrasive substances. The cover may be quickly taken off by loosening four screws and this exposes the air passage for inspection and cleaning when necessary.



Enclosed motor with section cut away to show details of construction. The housing is pressed steel

New Drive Chain Is Alloy Steel

DEVELOPMENT of a new steel drive chain which will be known as the "Hyper chain," has been announced by Link-Belt Co., Indianapolis. The chain is made from alloy steel, heat treated, and uses a new type of pin and cotter. Pins, bushings and holes in side bars are accurately ground and cotters are of a special type designed as not to work loose when once inserted and swelled into the holes provided in the pins for their reception.



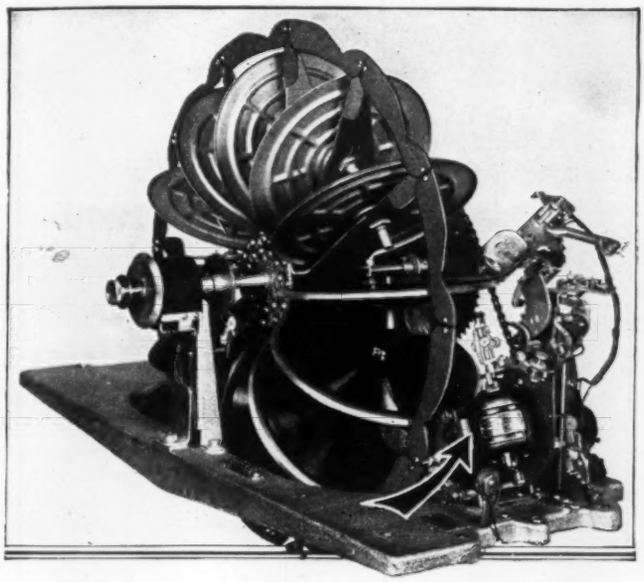
Parts of new alloy steel drive chain shown before and after assembly

The new chain combines great strength with durability. The "Hyper SS-40" chain has an ultimate strength of 75,000 pounds, and the "Hyper SS-124" chain, an ultimate strength of 150,000 pounds. Manufacturing tolerances to which "Hyper" parts are held are extremely fine, .001 inch, which insures press fits of the highest order. The chain was developed to meet a growing need for a stronger and more durable type of chain for heavy positive power transmission.

New Type Clutch Has Many Uses

CONSISTING of a central spindle carrying three pivoted arms at the outer ends of which are shoes fitting in an annular groove, a

HOW BODINE MOTORS SERVE MILLS NOVELTY AUTOMATIC PHONOGRAPHS



TO PAY the fiddler, the modern way, you merely slip a coin into a Mills Automatic Phonograph and a dozen different orchestras are awaiting their turn to entertain you.

Twelve records on individual turntables are mounted in a revolving Ferris wheel. When you insert a coin, a Bodine Type CR-3 Universal Motor indexes the desired record in line with the electric pickup. Then another Bodine motor, a Type CA-2 Shaded Pole Single Phase Motor with a special shaft extension comes into action and revolves the record by friction drive at proper speed. Instantly, the electric pickup communicates its vibrations to the dynamic loud speaker and the room is filled with stirring music.

From coast to coast these automatic phonographs perform under all conditions of weather and temperature. Reliability is paramount. Hence, Bodine motors are standard equipment on these instruments. Where unfailing performance is imperative, you will find Bodine motors.

IF YOU USE SMALL MOTORS—

let Bodine engineers test your motor-driven machines and engineer a motor to exactly meet your requirements. You will be surprised at the improved performance and often reduced manufacturing cost of your units if you use a motor that is designed for the job.

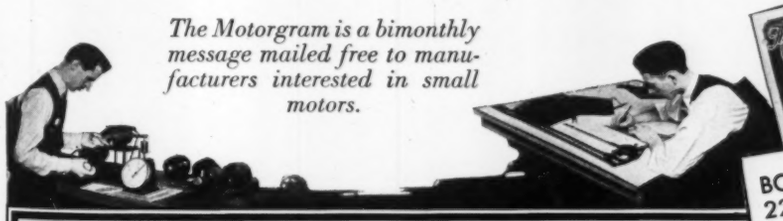
Bodine motors are available at all standard speeds in ratings from 1-1300 H.P. to 1-4 H.P. Send for the Bodine catalog. It is a valuable reference book on fractional horsepower motors for designers and engineers.

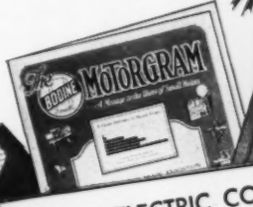
MAIL THE COUPON FOR BODINE CATALOG

BODINE

Bodine A.C. and D.C. Motors are interchangeable for dimensions and speeds..

The Motorgram is a bimonthly message mailed free to manufacturers interested in small motors.





BODINE MOTORS
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BODINE ELECTRIC CO.
2258 W. Ohio Street, Chicago
Please send items checked below:
☐ Bodine Catalog ☐ Motorgram
Name----------
Title----------
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"Pitter" one-way clutch is being marketed by the Universal Gear Corp., Chicago. The shoes are constantly kept in gripping position by light coil springs which, however, take no part in the drive.

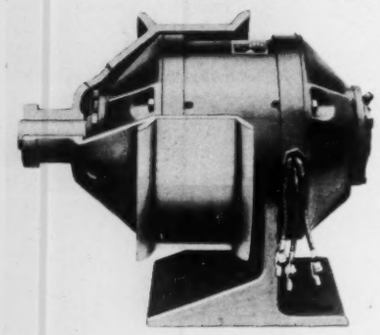
Constructional details of the clutch are shown in the accompanying illustration and additional description appears on page 22 of this issue. Shoes and annular grooves are made of high

View of clutch showing the central spindle carrying three pivoted arms equipped with shoes which fit in an annular groove

grade steel hardened and ground. The clutches are made oil-tight and filled with lubricant at the factory. The rapidity of action, it is claimed, is practically unlimited and with some now in use the shoes lock and unlock thirty and more times per second. Their broad range of application includes cutoff machines, load brakes on excavating machines, oil pumps, nail making and wire drawing units, punch and printing presses, automobiles, etc.

Motors Designed for Conveyors

SPECIAL motors designated Type M for runout tables in steel and tube mills and for conveyor work have been developed and are being built by the Reliance Electric & Engineering Co., Cleveland. These units are provided with a single heavy welded steel support located to permit the



Single heavy welded steel support is provided to permit rotation of pulley around motor

rotation of a pulley-type roller keyed to the shaft, around the body of the motor.

The motors with rollers are lined in tandem, the distance between being governed by the material conveyed. An installation may require any number of units depending on the distance of travel and roller spacing. Being fully enclosed, the motors are protected from injurious elements and insulation of asbestos and mica with which they are provided, will withstand extremely high temperatures. Large ball bearings carry the motor rotor and load.

Switching Action Features New Unit

OUTSTANDING among the features of the new "Noark Quadbreak" switch recently announce by Colt's Patent Fire Arms Mfg. Co., Hartford, Conn., is the switching action. It is this action which makes possible the four breaks in each pole, and is accomplished by two contacts fastened in an insulating slide which is guided edgewise between two insulating blocks and moves between two sets of clips. When the switch is thrown "On," the contacts are moved in between

Interior of switch showing insulating slide and clips

the clips and the current passed through, thus completing the circuit.

When the switch is thrown "Off," the contacts are withdrawn from between the clips and the insulating slide moved between, thereby acting as a constant barrier. This switching arrangement insures long life to the contacts, it is claimed, because of the absence of heating or destructive arcing when the switch is opened or closed. Combined with the sliding member and its movable contacts is the lower base for the stationary clips and the fuse block. The cabinet is stamped from steel and finished in enamel.

Announces New Motor-Driven Pump

CAPABLE of developing 39 pounds pressure per square inch, and able to pump 90 feet high, a new 2-stage pump has been added to a line of ver-

Standardize on

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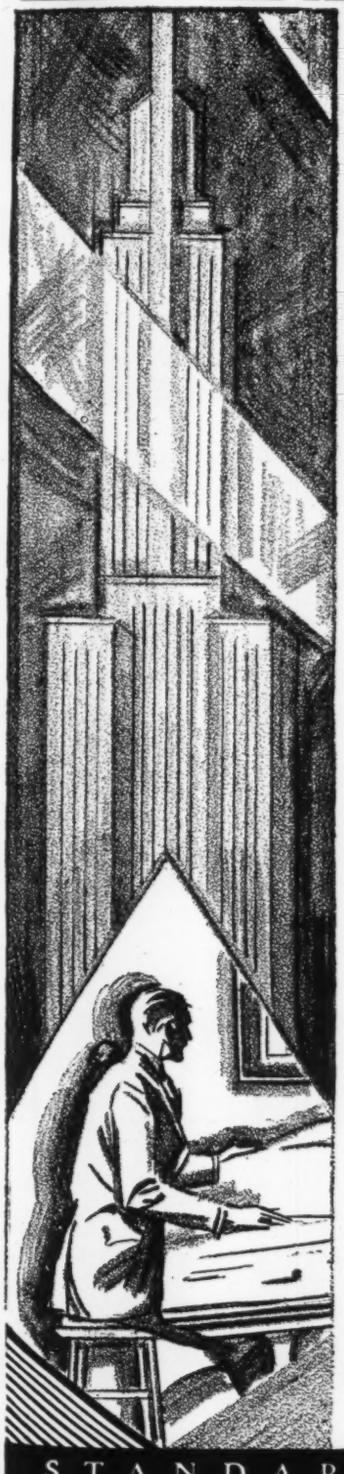
DRAFTING & SURVEYING SUPPLIES



DIETZGEN today has a most complete and standard line of drafting and surveying instruments, supplies and furniture—superlative quality blue print paper to meet every requirement of architectural, professional, industrial or school drafting. The name DIETZGEN is a reliable guarantee of quality and satisfaction.

Our widespread distributing and service organization bears evidence of the success of the policy of ever being on the alert, constantly experimenting in research and design, and ever seeking new and better methods of producing finer products.

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Our Prompt Dispatch Service, good dependable quality, complete standard line and popular prices induce many to use DIETZGEN'S catalog exclusively for their drafting and surveying requirements.

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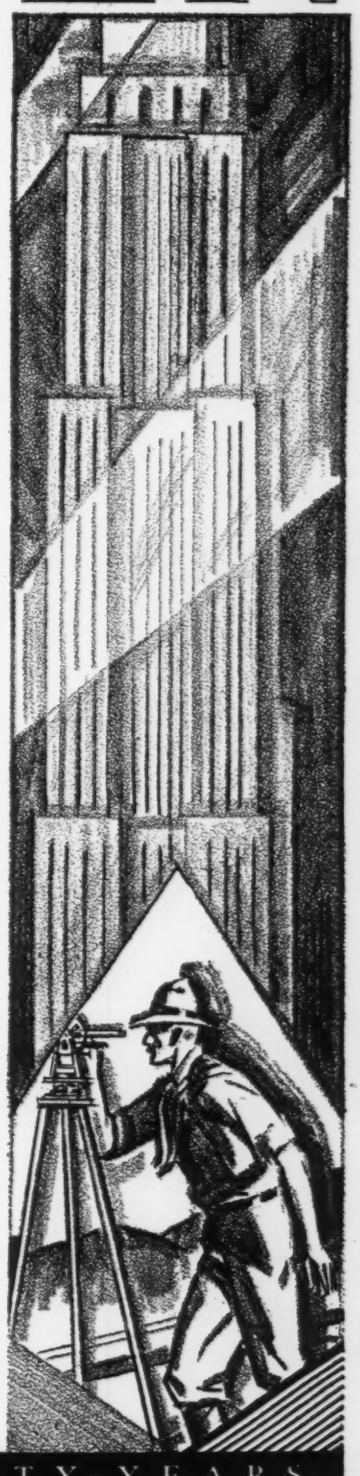

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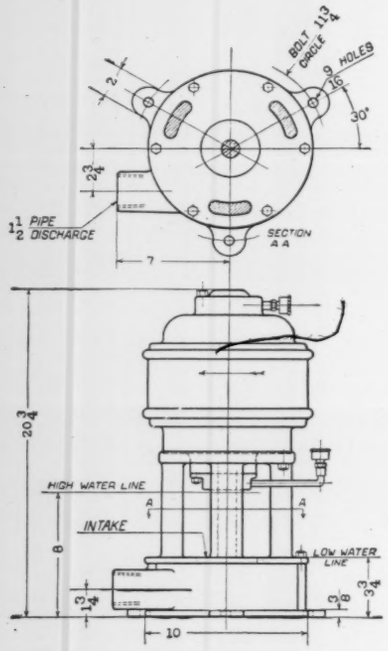
Manufacturers of Drafting and Surveying Supplies

STANDARD FOR NEARLY FIFTY YEARS

tical motor-driven pumps by Ruthman Machinery Co., Cincinnati. The pump, shown in the accompanying illustration, is equipped with a 1½-horsepower motor. It has no packing, thereby eliminating unnecessary friction, and permitting the full benefit of the ball bearing construction to be applied with full utilization of the power consumed in operation.

The intake is constructed with a stationary device of three curved blades to prevent the water from whirling and forming a conical shape. This feature prevents the entrance of air, making the flow solid and free from air bubbles. This pump has three distinct uses: First, for creating pressure for hydraulic movements; second, for supplying coolant to a number of machines from a central reservoir; and third for use with air cleaning and humidifying systems.

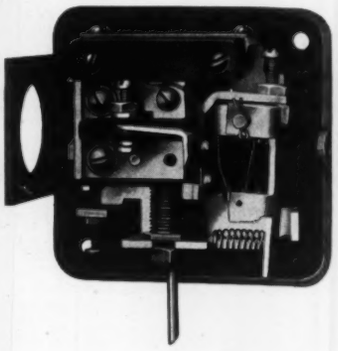


The intake of the 2-stage pump has a stationary device employing three curved blades to prevent the water from whirling and forming a conical shape, thereby keeping out air and making the flow solid and free from bubbles. No packing is employed

Thermostat Made for Three Ratings

WHERE close control of the heat is not required, the General Electric Company announces the CR-2992-A-1 thermostat for use with industrial heating units. The new device is available in three principal ratings, and the operating ranges are: 60 to 200 deg. Fahr.; 150 to 300 deg. Fahr., and 250 to 400 deg. Fahr. Current-carrying capacity is 15 amperes at 115 or 230 volt alternating current, or ¼ ampere at the same voltages direct current. The thermostat will operate with a differential of 5 per cent of the maximum temperature range.

The unit is mounted on a pressed-steel base which has two mounting holes and an upturned ear with an opening for a ¾-inch conduit. A drawn shell cover, held in place by two screws, completes the enclosure of the switch. Cover



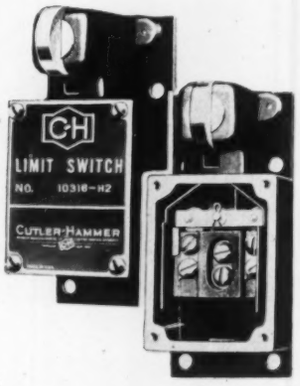
All steel parts of the thermostat are cadmium plated. A drawn shell cover encloses the unit

and base are finished in black duco. Clearance between all live parts and ground are in accordance with underwriters' specifications. All steel parts are cadmium plated to resist corrosion. The silver contacts afford long wearing qualities. The capillary tube consists of copper tubing of ⅛-inch outside diameter, having the bellows sealed on the switch end and the bulb on the immersion end.

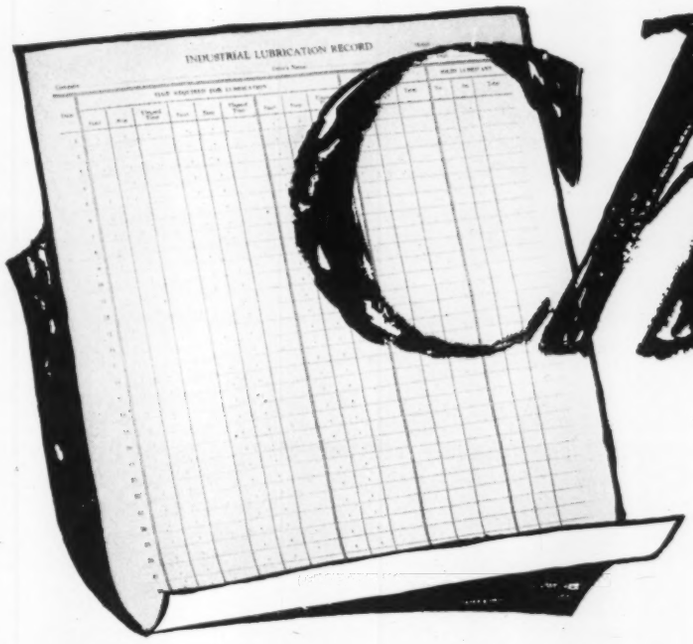
Some suggested applications of the new thermostat include wax pots on shoe machinery, laundry machinery, pressing and cleaning machinery, etc.

Switch Available in Two Types

A NEW, single-pole limit switch for general service is announced by Cutler-Hammer Inc., 5-12th street, Milwaukee. A small, exceptionally sturdy device, it can be applied on machine tools, conveyors, dough mixers, wash machines and other small equipment, to start, stop or reverse the motor at its limits of travel. Two types are furnished; one with normally closed contacts which are opened when the operating



Single pole limit switch provided with double-break contacts with silver rivet contact tips. Two types are available



Check up

Every phase of lubrication with these FREE COST SHEETS

THE time is long since over for Alemite Corporation to argue the advantages of Alemite High Pressure Lubrication Systems with the designers of American machinery.

Today the greatest proof of Alemite's sincerity in its desire to serve the designer lies in its offer to prove in your own plant on the machinery you build, the manifold savings effected by its systems.

Alemite's Free Lubrication Cost Sheets are designed to allow you to form an impartial estimate in your own shops of the increased efficiency and lubrication savings now possible.

The cost of a thorough lubrication test is small and the advantages you will discover are great. Select only one type of machine that you build. Install Alemite fittings and test this machine against another in the same service for sixty or ninety days.

You will find savings in every phase of this test with Alemite equipment. You will find that the Alemite fitted machine operates with a degree of production efficiency that approaches the ideal.

And you will discover the basis for Alemite's claim that Alemite High Pressure Lubrication Systems will pay their own cost and a dividend of 200% thereon to any plant the first year they are installed!

But—conduct your own test! After all, you are the man that Alemite must convince solely on the basis of your own findings.

ALEMITE
Industrial Lubrication

Alemite Corporation (Division of Stewart-Warner), 2644 N. Crawford Avenue, Chicago, Ill.

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Please send me Free Lubrication Cost Sheets.

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City ..... State .....

231

Above: Small pads for absorbing shocks in machines.
Below: Pad slotted to withstand great compression.

Elastic pads like these made of LINATEX when introduced into

MACHINE DESIGN

produce long life and quiet. We make all shapes to customers' requirements. They are the friends of careful machine designers.

LINATEX is *most resilient* and *does not harden nor crack* with age and use.

LINATEX may be new to you but it is being used extensively and has proven much more durable and economical than ordinary rubber. It is being used as illustrated above and in many other ways.

The large pad shown has borne a load of 160,000 pounds (2000 pounds per square inch) and is neither bent nor broken.

Write for descriptive catalogs and samples.

WILKINSON
PROCESS RUBBER SALES CORP.
53 West Jackson Blvd. Chicago, Ill.
Telephone Harrison 1116
Telegrams Wilprosaco

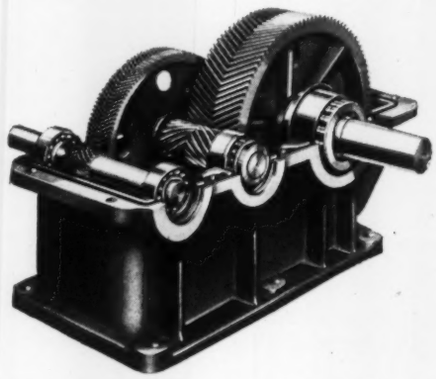
roller is depressed and the other with normally open contacts which are closed when the roller is depressed.

This new limit switch is 4-7/16 inches high, is compact and well built. Double-break contacts with silver-rivet contact tips reduce pitting and insure long life. The movable contact is connected to the machined steel operating roller through a sliding shaft and a heavy bakelite strip. A knockout hole in the end of the cast-bronze enclosing case and accessible screw terminals facilitate wiring.

Designs New Line of Speed Reducers

A NEW line of antifriction speed reducers known as "IXL Titans," has been announced by Foote Bros. Gear & Machine Co., Chicago. Available with either herringbone or helical gears, the units have many features in common. Cases are cast from close grained gray iron, treated to prevent flaking, and ribs are provided for bearing rigidity. An open view of type DB double reduction antifriction herringbone reducer is shown in the accompanying illustration.

Oil return grooves are provided for returning the lubricant to the bottom of the case. Splash lubrication is standard. The reducers are made in both single and double reduction types, the

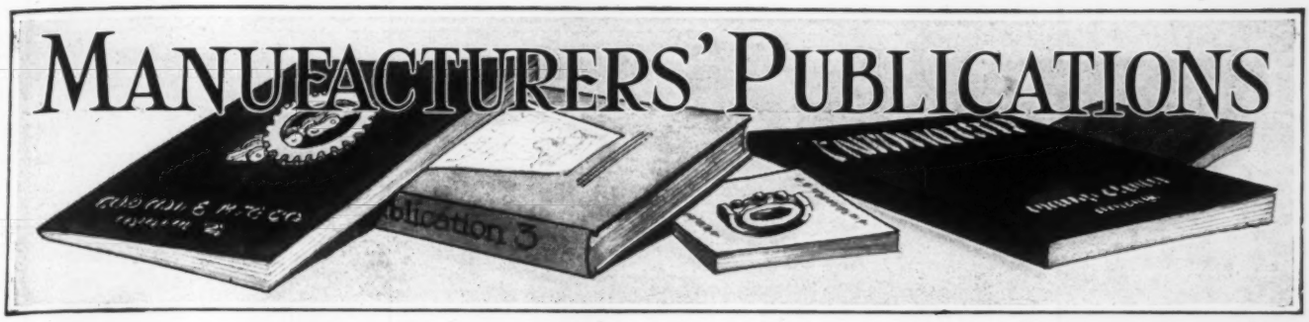


Type DB double reduction antifriction herringbone speed reducer

herringbone being designated as "SB" and "DB" and the helical as "SX" and "DX." On the single reduction types the ratios range from 2.87 to 9.9 and on double reduction units 9.95 to 96.2.

Chrome nickel steel in place of carbon steel now is being used for the fingers of an improved clutch manufactured by Carlyle Johnson Machine Co., Manchester, Conn. A full description appeared in the May issue but the title was misleading in that reference was made to carbon steel fingers. However, the text of the article was correct.

MANUFACTURERS' PUBLICATIONS



Publications listed in this section may be obtained without charge from the manufacturers of the products or through MACHINE DESIGN.

ALLOY STEELS—International Nickel Co. Inc., New York, is distributing a booklet covering alloy steels in the railroad field, by Charles McKnight, who is in charge of alloy steel development for the company. It discusses increase in strength with decrease in weight attained by use of alloy steel, comparative physical properties of alloy and carbon steels and where each is economical.

FLEXIBLE SHAFTS—Engineering data covering its flexible shafts are given in a 16-page folder designated bulletin E-125 being distributed by the Stow Mfg. Co. Inc., Birmingham, N. Y. Various applications are given and illustrations show a number of uses for this type of shafting.

BEARING SEAL—Armstrong Cork Co., Lancaster, Pa., is distributing a leaflet describing its cork for sealing antifriction bearings. The publication shows why cork has found wide acceptance for this purpose and indicates briefly how to use it to obtain best results. Diagrams and illustrations supplement the text matter.

LUBRICATION—Special problems of lubrication for various types of machinery are considered in the monthly publication on lubrication by the Texas Co., New York. The current issue is devoted to lubrication of highway construction machinery. Illustrations show numerous types of this class of equipment and the means taken for efficient lubrication.

ROLLER CHAIN DRIVES—Link-Belt Co., Indianapolis, recently issued a 47-page booklet announcing its stock line of roller chain drives ranging from ½ to 100-horsepower and ratios 1 to 1 and 6¼ to 1. Features of the roller chain are summarized and selection data from which designers may choose the type of drive to meet their conditions are given. Numerous illustrations show typical installations.

MOTORS—Across-the-line motors manufactured by the Reliance Electric & Engineering Co., Cleveland, are described in a bulletin now being distributed. The motors are equipped with ball bearings and are adapted to two and three-phase alternating-current circuits. A series of questions and answers presents the advantages of this class of equipment. A table of ratings and a diagram of torque characteristics convey added information.

ELECTRICAL APPLIANCES—General Electric Co., Schenectady, N. Y., has issued a series of bulletins for its data book covering strip heaters, capacitor motors, oil circuit breakers, brush-contact switches, air circuit breakers, outdoor and automatic switching equipment, engine-temperature indicator, enclosed starting rheostats, hook-operated disconnecting switches, fabroil gears, textolite gears, and vertical induction motors. Illustrations and data present a comprehensive description of the different units.

STEEL CASTINGS—Industrial Steel Casting Co., Toledo, O., sets forth in a bulletin the facilities it has for making steel castings for special purposes.

SPEED REDUCERS—Cullman Wheel Co., Chicago, describes in a bulletin four sizes of its single type speed reducers and three sizes of double type with ratios of reduction for each size. Typical installations are shown.

SWITCHES—Bulletin 830 recently issued by Allen-Bradley Co., Milwaukee, describes a newly developed automatic pressure switch and accessories designed to be used in connection with automatic motor control for compressors, pumps and other equipment not exceeding 150 pounds pressure per square inch.

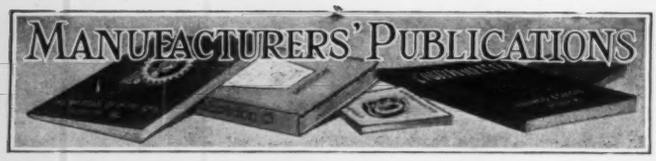
NITRIDING—A bulletin by the Leeds & Northrup Co., Philadelphia, deals with the homo method of nitriding of steel. It describes the company's furnace designed for this process, shows typical installations, a diagram of the furnace, its operation and advantages of its use. It is illustrated.

RECORDING GAGES—Bristol Co., Waterbury, Conn., is distributing a new catalog of its recording pressure and vacuum gages. Illustrations present many types of gages and charts for industrial purposes. Various accessories are shown. Accompanying the catalog is a list of charts available for use with the various types of recorders.

SPECIAL STEELS—Edgar Allen & Co., Sheffield, Eng., has issued a booklet on its malleable stainless steels, showing method of production and various uses to which it is put. It also has issued a new edition of its booklet on carbon tool steels with all information revised and carefully checked. It constitutes a handbook for this class of steels.

CENTRIFUGAL PUMPS—Barrett, Haentjens & Co., Hazelton, Pa., manufacturer of centrifugal pumps, has published a series of bulletins on effective means for priming centrifugal pumps manually. The bulletins cover the theory of priming, submerged pumps, use of foot valves, air, steam and water ejectors, hand-operated and motor-driven priming pumps and arrangement of intake pipes.

FORGINGS—Heppenstall Co., Pittsburgh, presents in a current bulletin the subject of steel forgings as produced in its plants. Manufacture, from the ingot to the completed forging, is illustrated and many types of



products are shown. Tables are given of weights of square and round steel bars and circumferences and areas of circles. A table converting millimeters into inches is included.

TACHOMETER—O. Zernickow Co., New York, is distributing a leaflet on its anemo tachometer, an air velocity meter. Its applications in industry are outlined and construction and operation are described.

ARC WELDING—Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa., continuing its series of bulletins on arc welding, has issued numbers 12 and 13. They present late developments of welding technique of interest to designers and users of welding and welded products.

STEEL CASTINGS—Lebanon Steel Foundry, Lebanon, Pa., manufacturer of steel castings, devotes a current bulletin to presentation of strength of its castings for use in cranes and hoists for industrial plants. Illustrations supplement the text.

ELECTRIC MOTORS—Howell Electric Motors Co., Howell, Mich., manufacturer of electric motors, is distributing a bulletin on its single phase motors. It discusses needs of variation in type of motors for various purposes and offers service to obtain the proper type for a given use.

ALLOY STEEL CASTINGS—Michigan Casting Co., Detroit, is distributing a number of bulletins on its alloy steel castings for a wide variety of purposes. Illustrations indicate the applications of its products and a chart shows the physical properties of the alloy, which is designed especially for use under high temperature conditions.

COPPER GROUNDS—Copperweld Steel Co., Glassport, Pa., has initiated publication of a bimonthly bulletin on methods of electric grounding by use of its ground rods.

SEAMLESS TUBING—Sumerill Tubing Co., Bridgeport, Pa., tells in a leaflet how seamless steel tubing is drawn. An illustration shows a miniature draw bench and the text describes its operation for production of tubing.

COMPARATORS—Jones & Lamson Machine Co., Springfield, Vt., features in a current bulletin the accomplishments of its comparator for testing the variations of a screw thread, giving illustrations of true and faulty threads as shown on the screen of this instrument.

GRAPHIC TIME RECORDERS—Esterline-Angus Co., Indianapolis, describes in a current bulletin some of the uses to which its graphic time recorders can be put to give a permanent record of operations and events in industrial and other operations. The workings of the recorders are illustrated and their construction given in detail.

WELDED CHAIN—Facts and figures including sizes, weights, loads and directions for taking measurements of all types of welded chain are given in a new booklet recently issued by American Chain Co. Inc., Bridgeport, Conn. Information also is presented on how to order in accordance with the A. S. T. M. and A. R. A. specifications, as well as rules and cautions issued by the chain institute.

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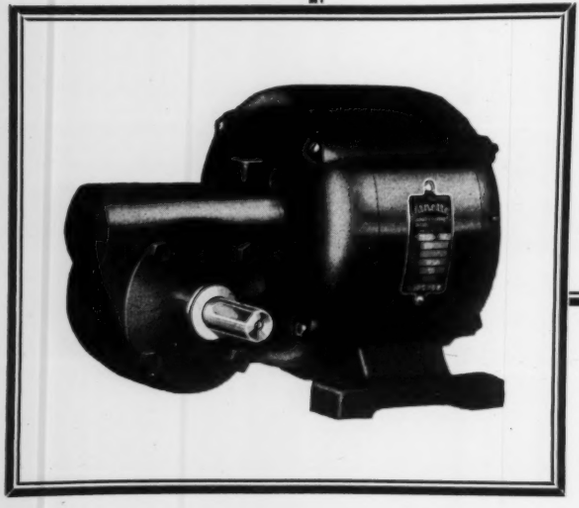

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Write for Bulletin SR-529

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